MACHINERY.

June, 1905.

HIGH-SPEED PLANERS.

A DISCUSSION OF THE MOST DIFFICULT PROBLEM NOW BEFORE MACHINE TOOL DESIGNERS—THAT OF ADAPTING PLANERS TO THE USE OF HIGH-SPEED STEEL.

A. L. DE LEEUW.

A. L. DE LEEUW.

With the advent of high-speed tool steel, the demand has come for high-speed machine tools. This demand could easily be satisfied in a great many cases. The ordinary lathe, for instance, was generally run at such low speeds that it was possible to speed up the countershaft, thus preserving the same pull on a piece of given diameter; or, in other words, raising the amount of power consumed by the lathe. Boring mills, boring machines and the like, could be treated in a similar manner. In many cases this was not quite sufficient; the machine was not up to what the tools would stand, and as a consequence machine-tool builders got out what they called their rapid reduction, or rapid production, lathes and boring mills. In these machines, the horse-power consumed by the machine was raised by increasing belt width and speed, and by strengthening the gearing, and in most cases also the frame work. This is all very simple and, as one prominent tool builder expressed it, there is no reason why a lathe could not be designed which would take a cut a foot deep, with a foot of feed, and running a mile a minute, provided the tools would stand such a cut—and, provided somebody could be found willing to pay for it. This statement may be slightly exaggerated but it expresses the fact fairly well, that limits in that direction are far away and not yet reached by a good deal. All the tools mentioned are machines in which the tool is all the time in contact with the work; they are rotary machines. An entirely different problem presents itself, however, when an attempt is made to increase the speed of reciprocating machine tools, such as planers, shapers and slotters. The planer problem has been attacked a number of times and by a number of people, but with indifferent result, so far. It may be well, therefore, to look into the matter, and try to find out what are the essential requirements of a high-speed planer, what difficulties one meets, and finally, as a result, how to overcome them.

For a long time all efforts were directed toward increasing the return or backing speed. Return speed was the great selling point of planers. This is the case even now, though I fear, not deservedly so. Let us take a planer running at a cutting speed of 20 feet per minute, and having a return speed of 80 feet per minute. Let us try to find out how much is gained if this return speed is increased to 100 feet. Suppose a cut 400 feet long has to be taken. This does not mean that the piece of work is 400 feet long but that the total of a certain number of cuts is 400 feet. The piece may be 10 feet long, and may require 40 cuts, which is equivalent to one cut 400 feet in length. The time for the cut is $400/20 = 20$ minutes. The time for the return is $400/80 = 5$ minutes. Total 25 minutes. If the return speed is increased to 100 feet, then the time for backing is $400/100 = 4$ minutes; while the time for cutting is still 20 minutes; making a total of 24 minutes. This shows that by increasing the return speed from 80 feet to 100 feet, the time saving is one minute out of every 25 minutes, or 4 per cent. This figure is a little bit better than the actual showing would be, for the time of reversing—that is, for slowing down and speeding up again—has not been considered. In order to get this increased speed certain things have to be done which cost money and, worse than this, become a source of weakness to the planer. The countershaft has to be speeded up, or the belt width must be increased so as to get along with less gear reduction, or planer shafts must be run at a higher speed, causing increased wear, and liability of sticking. In short, it becomes a question whether the "game is worth the candle." It is doubtful in my mind whether 100 feet return will make a better showing at the end of the year than the 80 feet.

The amount of brains, energy, skill and experience brought to bear on this question, makes me think of hunting a rabbit with a six-inch quick-firing gun. Of course, this increased backing speed was the only game in the woods up to the time when high-speed tools made their appearance. It is different now. Both cutting and return speeds are taken in hand. Not only is the cutting speed increased, but a number of attempts have been made to get the benefit of various cutting speeds, to suit various conditions. A planer may be called upon to work on different kinds of materials, of all hardnesses, and with tools of all kinds of stand-up qualities. All sizes of cuts may have to be taken; all kinds of finished surfaces may have to be produced. The work itself may be of all grades of stiffness, and of any weight from the smallest piece up to 50 or 60 tons. All these different conditions call for a planer which should be flexible. It should be possible to plane slow or fast, and to return fast or faster. It should be possible to do this without spending much time in changing the drive. To show why all this is necessary, I will give here a few examples of conditions which may prevail, and of the way the planer should run to meet those conditions.

I. The work is a heavy cast-iron piece of considerable hardness, and with plenty of stock to remove.

In this case, the planer should run fairly slow on the cut, say about 20 or 24 feet, and take heavy cuts. A tool will stand such heavy cuts at low speeds better than when running at high speed with light cuts. The amount of metal removed per hour will be as much or more. Besides, the table and the work having to travel a lesser distance for the removal of a given quantity of stock, the waste of power (used merely to move table, load and mechanism) is less. The wear on the planer itself is also less. It should never be forgotten that the amount of metal removed is the object. The return speed should be moderate, as the quick starting and moving of a heavy load is liable to strain the planer mechanism severely.

II. The work is heavy, but has relatively little stock to remove.

In this case, the cutting speed may be as high as the tool will stand, but the return speed should be kept within moderate limits, and for the same reason as under I.

III. The work is of light weight, but stocky, and much metal has to be removed.

In this case, the cutting speed should be moderate, but the return speed may be high.

IV. The work is light, and has little stock to be removed.

In this case, both cutting and return speeds can be high.

There are several other things one should observe, however,

ADOLPH L. DE LEEUW was born in Holland in 1861. He was educated in the University of Leyden, from where he graduated in 1884 as bachelor of science and also as mechanical and electrical engineer. He has held the positions of draftsman, designer in charge of construction of special machinery and chief designing engineer for the following concerns: The Pennsylvania Railroad Co., the Morgan Engineering Co., the Springfield Machine Tool Co., the Warder, Bushnell & Glessner Co., the Garvin Machine Co., the Pond Machine Tool Co., and the Niles Tool Works Co. His specialty is machine tools, both large and small, and automatic machinery. Besides his contribution to MACHINERY Mr. De Leeuw has written for the *American Machinist* and the *Railroad Gazette*.

if one wants the best results. For instance, it is of little benefit to have a high return speed on short stroke. This simply results in wearing the planer without accomplishing any good. Again, where the work is steel, a fairly high cutting speed may be used, even if there is much metal to be removed; for most high-speed steels work better at a high speed on steel than on cast-iron; so much so, that some tool steels do not show up well at all when cutting steel unless run at a decidedly high speed. Where brass is to be planed, high cutting speed may be used; but with some bronzes, a low cutting speed is absolutely necessary, if one does not want to tear the work. It is plain, therefore, that a machine which offers only one cutting speed and one return speed, is far from approaching the ideal construction.

In addition to all this, there is something else to be considered, which tends to make the problem still more complicated; it is that, though the tool may stand a certain speed when cutting, it will not stand this speed when entering the work. It is well known by this time that most high-speed steels stand shock but poorly, and for this reason cutting speeds of planers are frequently kept down, though all other conditions may be favorable for high speed.

In order to get a clear understanding of the difficulties one meets in trying to solve the planer problem, I will divide planers in three groups, viz.: Belt-driven planers, motor-driven planers, with some auxiliary mechanism between the motor and the machine proper, and direct-connected motor-driven planers. The first kind, being the oldest, ought to be treated respectfully; but it is not likely to get the job. The belt-driven planer has some inherent limitations which defy ingenuity. Its most troublesome weakness lies in the fact that the amount of horse-power taken by the planer depends on belt width and belt speed. To increase the width means increased difficulties of shifting, increased pressure on the bearings and increased weight of the pulleys. To increase the belt speed means increased tendency to slip, increased shaft speed, and with it, increased difficulties of lubrication. And after all, there are certain limits even if one is willing and able to overcome the above-mentioned difficulties. Belt width cannot be increased indefinitely and neither can belt speed. To illustrate, I will take for example some planer running the old-fashioned way, and see what must be done to double its speed, both for cutting and for return.

Belt-driven Planers.

Suppose the cutting speed to be 20 feet, and the return speed 60 feet. Suppose further, the cutting pulley to be 30 inches in diameter for 4-inch belt, and the return pulley to be 24 inches diameter for the same width of belt. The belt is supposed to travel 50 feet for one foot table travel, which would give a pull at the table rack of 50x4x50 = 10,000 pounds, figuring on 50 pounds pull per inch width of belt. The cutting pulley will have to run 128 R. P. M. for 20-foot table speed; and the return pulley will have to run 384 R. P. M. for 60 feet return. The speed of the cutting belt is 50x20 feet = 1,000 feet, and the speed of the return belt is 2,400 feet.

If we now go to work and double the speed of the countershaft, so as to get 40 and 120 feet speed of the table, we will have to increase the speed of the cutting belt to 2,000 feet per minute. The return belt will run 4,800 feet; and when the return belt is on the tight pulley the tight cutting pulley will have a circumferential speed of 30/24x4,800 feet = 6,000 feet. A speed of 4,800 feet of the belt is possible, though not desirable. By using pneumatic pulleys (pulleys with perforated rim), slippage may be overcome, but the belt is liable to flap. A circumferential speed of 6,000 feet for a cast-iron pulley is unsafe and, generally speaking, steel-rimmed pulleys have not enough side stiffness for a shifting belt.

When returning, the pulley shaft runs 768 R. P. M. Though this speed is not at all phenomenal, it is an awkward speed for good lubrication, especially with a belt tugging at a shaft. Good designing may overcome this difficulty, but it is a difficulty nevertheless. There is, however, another difficulty which cannot so easily be swept out of the way. Increasing the speed of the pulleys increases the horse-power delivered by the belt, and also the momentum of the pulleys and other moving parts. The horse-power increases in direct ratio to the speed; but the momentum changes as the square of the speed. When shifting, therefore, four times the momentum of the parts must be stopped by double the belt power; so that, as will easily be seen, there is double the tendency to slip. To tighten the belts more is no solution for this would lead to other troubles. It would be necessary, therefore, to change all the leading dimensions given above; and even then, no set of dimensions will overcome all the difficulties at once. And, even though one may be satisfied with the mechanical arrangement of the planer, the result obtained is not at all what one should have; for there is only one cutting speed and one return speed. The cutting speed being high, the planer becomes useless for such work as requires slow cutting. The next step would be to use a variable speed countershaft, either by interposing a set of cone pulleys between the line and the counter, or by using one of the many variable-speed devices now on the market. However, if this is done the return speed goes down with the cutting speed, which, of course, is undesirable.

A number of planers have been furnished with a countershaft which gives a variable speed to the cutting pulley, while the return pulley runs at a constant speed. This object may be reached in different ways. One might have two countershafts; one with a variable-speed device for the cut, and the other a simple countershaft for the return. This is a better arrangement, though not at all free from objections. There are still the same limitations to high return speed, and, besides the return speed being constant, it must be chosen low enough for the most unfavorable conditions, that is, for heavy load and short stroke. One of the ways in which this arrangement is carried out is the Reeves variable planer countershaft. This arrangement, and many others of similar construction, is handy to operate and is a decided improvement over the old one-speed planer; but the peculiar conditions under which a planer works are hard on the countershaft, and it becomes necessary to rate its horse power low if one wants to keep the contrivance out of the hands of the doctor.

In order to get variable return as well as cutting speed, one might use two of these countershafts. This would solve the problem so far as speeds are concerned; but the other objections, enumerated above, would still exist. The countershaft might be driven by a motor; either placing the motor on the floor, against a wall or post, hanging it from the ceiling, or putting it on top of the planer; then belting over to the counter, or by gearing the counter to the motor. The problem remains the same, except that a new and troublesome element has been introduced, viz., the motor. In order to see what influence the motor has on the planer problem, we will have to study the reversal of the planer.

The Action during Reversal.

This reversal takes so little time that it is very hard to notice all that happens, by simply looking at it. A little reasoning is necessary to see it clearly. Suppose the planer is at the end of a cutting stroke. The dog strikes the tappet, and begins to move it. This starts the movement of the belt eye, guiding the cutting belt. This belt is moved from the cutting pulley onto the loose pulley. Meanwhile the planer keeps on moving, by the momentum of the moving parts, especially the pulleys. The pulleys are moving relatively slow; but even at this slow speed, they would be able to move the planer several feet if nothing else happened. Immediately after the cutting belt is moved onto the loose pulley, the belt-eye guiding the return belt, begins to move, and brings this belt onto the return pulley. This belt is running in opposition to the pulley and therefore has a braking effect as soon as it comes in contact with the pulley. The pulley slows down, and so does the planer. The movement of the belt-eye also becomes slow and, by the time the full width of the belt is on the return pulley, this pulley, the planer, and the belt-eye have come to a standstill. However, this is only momentary. The belt begins to move the pulley in the opposite direction. As long as the pulley has not acquired the full speed of the belt, there will be slippage. This slippage takes place, therefore, during all the time that the planer slows down and picks up speed again in the new direction. If the planer comes to a standstill, before the return belt is fully on the pulley, the act of shifting this belt comes to an end, and the planer has to start on

the return stroke with less than the full width of the belt on the pulley. This will actually happen when the speed of the cutting pulley (and therefore its momentum) is too low, and it will happen especially when there is a heavy load on the planer, or when the cut is taken up to the extreme end of the stroke, so that the momentum of the pulley has to overcome the resistance of the cut.

When the shifting is not complete, the belt is apt to suffer; moreover, the planer may come to an unexpected standstill, or drag along on the reverse stroke with continuous slippage. This is one reason why a low belt speed for the cut is not advisable; for, in that case, the momentum of the pulley is not enough to carry the shifting through to the end. When the reversing belt enters on the tight pulley, it must start this pulley, and bring it up to a high speed. The reverse pulley is generally smaller than the cutting pulley, so as to get sufficient speed without making the countershaft pulley too large. The belt, acting on this small pulley, must give momentum to a large one. This is the reason why cutting and return pulleys should be made as nearly equal in size as is practical.

Motor-driven Planers.

We have drifted away from the motor, but purposely, for we want to show what the motor will have to do. Suppose the motor is a shunt-wound machine. Such a machine tends to run at the same speed, regardless of load. If the armature shaft of a shunt-wound motor is clamped, so as to prevent it from turning, and current is allowed to flow through the armature, the amount of current will be limited only by the resistance of the copper bars (or wires) on the armature, and by the resistance of the wiring between the dynamo and the motor. This resistance is very low, and we will have a short-circuit. Leaving the resistance of the outside wiring out of consideration, the amount of current going through the stationary armature can be figured, if we know the resistance of the armature conductors. Suppose this resistance is 1/10 ohm (which is really higher than will be found in a good motor) and suppose the voltage to be 220; then 220x10 = 2,200 amperes will flow through the armature. This, of course, is sufficient to burn out the motor unless protected by a circuit-breaker or fuses. Suppose the normal load of the motor is 100 amperes; then, when these 100 amperes flow against a resistance of 1/10 ohm, 10 volts will be lost in the armature. There are still 210 volts left, to be counteracted by something else. This something is the counter-electromotive force generated by the armature when running at its proper speed. Suppose the motor runs 630 R. P. M.; then these 630 R. P. M. must generate an electromotive force of 210 volts. This is three turns per minute for every volt. If, for some reason or other, this speed drops, then the 210 volts will not be fully made up, and more amperes will flow through the armature. Suppose, for instance, the speed of the motor drops to 600 R. P. M. This is not a very great drop as far as speed goes, but the consequences would be very noticeable. This drop of 30 R. P. M. would cause a shortage in the counter-electromotive force of 10 volts, which means that so much additional current would have to flow through the armature, as to cause a drop of 10 volts by the resistance alone. As this resistance is 1/10 ohm, the current will have to be 100 amperes and the total amount of current flowing through the armature will be 200 amperes, or twice the full load current. As, in reality, the resistance of the armature is much less than was assumed here, the increase of current will be much greater than was figured. This increase, however, is great enough to cause harm to the motor if it takes place during a sufficiently long time, or if it is repeated frequently. This drop of speed is exactly what happens if the motor is geared or belted to a planer countershaft, and this is why adding a motor to a planer is adding trouble to trouble. The countershaft, being driven by the motor, tends to keep its speed. The lower pulleys *must* come to a standstill every time the planer reverses; and, though this standstill lasts only an instant, there is still to be considered the decrease and increase of speed before and after the standstill is reached. Countershaft and machine pulleys being connected by a belt, there will be a tug of war, every time the planer reverses, and the consequence is that the belt slips and the countershaft goes down in speed. This is the reason why an ammeter shows such antics when in circuit with the motor which drives a planer.

This momentary overload of the motor makes it necessary to use a much larger motor than is required to drive the planer, and take the cut, and even then conditions must be favorable to avoid overload and sparking. If the motor is on the floor, and belted to the countershaft, the bad results are materially diminished. In general, it is advisable to have as much belt as possible between the motor and the machine, as this allows of more stretch and slip, and thus reduces the slowing-down effect on the motor. The overloading of the motor is at its worst when it is geared direct to the countershaft. Various means have been suggested to relieve the motor; among them are compounding the motor, putting a flywheel on the armature shaft, and the combination of those two. As to compounding the motor, this actually relieves it; for a compound wound motor is more elastic as to speed. However, it has the disadvantage of speeding up when the load is light; that is, just before the cut starts. As a consequence, the tool hits the work at a high speed, and this, as we know, is very objectionable. Another point, though of minor importance, is this: Every motor makes a humming noise, some more than others. This noise is not noticed after a while, because the noise is of the same pitch and the same intensity all the time. A compound wound motor, however, slows down and speeds up again at every reversal. This change of speed changes the note of the motor, and one notices the noise as soon as one comes within hearing distance, even outside the shop. Of course, this objection is not enough to condemn the compound-wound motor, but it is certainly not a point in its favor.

Putting a flywheel on the armature shaft is done so as to carry the motor over the dead point, and to prevent it from slowing down, and thus taking an excessive amount of current. It does this, it is true, but it compels the belt to slip the full amount, and consequently, is hard on belts and bearings. The combination of compound motor and flywheel strikes me rather as a joke; the motor is compounded, to allow it to slow down, and the flywheel is put on to prevent it from doing this. I found that the best way of all, was to use a shunt-wound motor, and make it big enough; and further, to keep as many belts as possible between the motor and the machine, and also to make them as long as possible.

All things considered, driving a planer by hitching a motor to the countershaft is not much better than a makeshift, and it is not going to be the ultimate way, if I am any prophet at all.

Planers Operated by Clutches.

As the shifting of the belts constitutes a limit to the possibilities of a planer drive, it has long been recognized that it would be advisable to get along without them. This led to the clutch-driven planers. All kinds of clutches have been designed to do the trick, a few good, but most of them bad, worse, or impossible. It was readily recognized that positive clutches would not do, so the inventors and designers settled down to the friction clutch. It was not so readily seen, however, that the manner of throwing in the clutch, or rather, the method for getting pressure behind the clutch, was of only secondary importance. So far as I can see it makes very little difference whether the clutch is thrown in by air, electricity, a lever, or the sole of my foot; its limitations, and the troubles it causes, lie in something else. The whole story is this, a friction clutch slips, or else it does not slip. When it slips, it does not work; and when it does not slip, then it is a positive clutch, and therefore, no good for a planer. The ideal friction clutch for a planer should slip enough to pick up the load gradually, and yet quickly, and should not slip at all when taking the heaviest cuts the machine has to take. This sounds simple, but to make such a clutch is a different thing. There are at the present time only two styles of clutches which are considered for a planer drive: the magnetic, and the pneumatic clutch. The first one has been cursed so often and so heartily that I am afraid the bottomless pit is not deep enough for it. The pneumatic clutches have not reached this stage—yet. Whether they ever will, I am not prepared to say. But the chances are somewhat in favor of the pneumatic clutch, and this does not lie so much in its qualities, as in the fact

that the average tool builder, tool designer, tool user, and the operator know more about air than about electricity. It would require too much space to give a somewhat complete essay on magnetic clutches. It may be well, however, to point out why they have failed to come up to the high expectations they once raised.

A magnetic clutch consists of a pair of iron-clad electromagnets, and an armature, or keeper, which is attracted now by the one then by the other, according to which magnet receives the current. A simple reversing switch, operated by the dogs or by hand, controls the current. Nothing can be simpler. A magnetic clutch is a friction clutch pure and simple; the magnet turns free on the shaft, and is driven by a gear. The armature is keyed to the driven shaft, and magnetism simply serves to press one member against the other. Therefore, all the troubles the friction clutch is heir to, appear in the magnetic clutch. The sum and substance of these troubles was given in the sentence, "A friction clutch slips, or else it does not slip." It was soon found out, that a magnetic clutch slips, and this caused wear. Some makers drilled a large number of holes in the surface of the clutch, and filled these holes with cork plugs, which would last sometimes a whole day, though generally much less. Other makers substituted hardwood for the cork, and made the plugs smaller in number and larger in area.

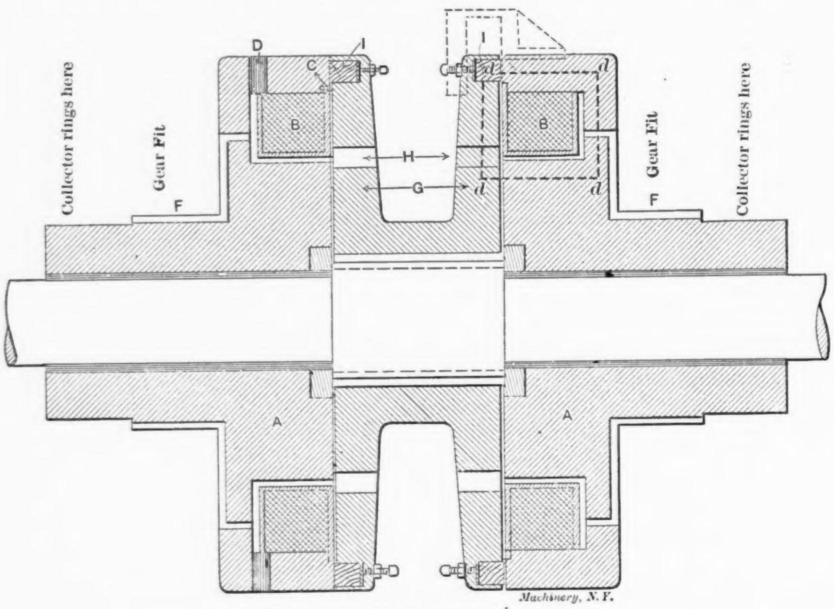


Fig. 1. Section of Magnetic Clutch.

Fig. 1 shows a section of a magnetic clutch and its armature. It is not drawn to scale, and should only be used as illustrating the principles of construction. *A* is a steel casting (the clutch), bored out to receive a bushing, which revolves on the shaft. It contains a coil of insulated copper wire, *B*. This coil is placed in a brass cage, so as to be easily inserted and removed. Space is left between the steel casting and the brass cage, for the purpose of ventilation. Ribs on the brass cage prevent it from shaking, while a number of screws, *C*, hold the cage in place. The annular space around the cage is connected with the atmosphere by a number of holes, *D*, drilled through the steel casting. The leads are carried off through channels, and are connected to the collector rings. Brushes deliver the current to these collector rings. The part, *F*, of the clutch is turned up for a gear fit, and is provided with a keyway. The armature is a hub, with a couple of flanges, one for each clutch. The flanges, *G*, are provided with a number of holes, *H*, for ventilation, and a number of other holes, *I*, into which hardwood blocks are fitted. A washer and setscrew behind each block serves to set them up, so that all project an equal distance above the flange. The path of the magnetic circuit is indicated by the dotted line *d-d-d-d*. The wood blocks serve two distinct purposes. In the first place, and as was mentioned, they provide the wearing surface; it is for this purpose that they are made adjustable. But there is another purpose, and an entirely different one. If the current is turned off in an electromagnet, the iron will lose its magnetism, but not immediately. At first the magnetism diminishes very rapidly, but this drop becomes slower and slower, and an appreciable time goes by before the magnet has lost practically all its power. This phenomenon is much more marked when the armature is in contact with the magnet. It is called residual magnetism. This residual magnetism is the bugbear for the designer of a magnetic clutch.

Suggestions in Regard to the Design and Operation of Magnetic Clutches.

Fig. 2 shows a pair of clutches with their armature in diagrammatic form. The armature is supposed to be in metallic contact with clutch *A*, while there is supposed to be an air gap of one-eighth inch between it and clutch *B*. It is well known that very little current is required to force a large amount of magnetism (or, as it is commonly expressed, a great number of lines of force) through a circuit of iron or steel, and that a great many times that amount of current is required to force the same number of lines through the same length of any other material as, for instance, air, or water, or brass. How many times depends on how nearly saturation is reached. Under certain conditions this number may be 3,000, for example. This means that the same amount of current which will force a certain number of lines through 30 inches of steel, will force the same number of lines through 1/100 inch of air, or any other non-magnetic material. A very little residual magnetism in clutch *A*, therefore, is capable of preventing clutch *B* from pulling the armature over to its side, even if *B* is magnetized the full amount. The wood plugs, mentioned above, serve the purpose of keeping armature and clutch separated, so as to keep the residual magnetism down to a minimum. This they do, but at the expense of the efficiency of the clutch. As explained, the existence of an air gap between clutch and armature makes it necessary to employ a very much heavier current in order to get the same amount of pull. It was found that about 3/64-inch air gap was necessary on a 33-inch clutch. Not only does the air gap allow the magnetism to discharge more quickly, but whatever magnetism is left has not the same holding-on power as when the clutch and armature are metal to metal. Even with the air gap, an appreciable amount of time lapses before the other side of the clutch gets power enough to pull the armature over to its side. This is due to the fact that a high degree of saturation of the clutch is used, and that a piece of iron or steel requires some time before it reaches this state of saturation.

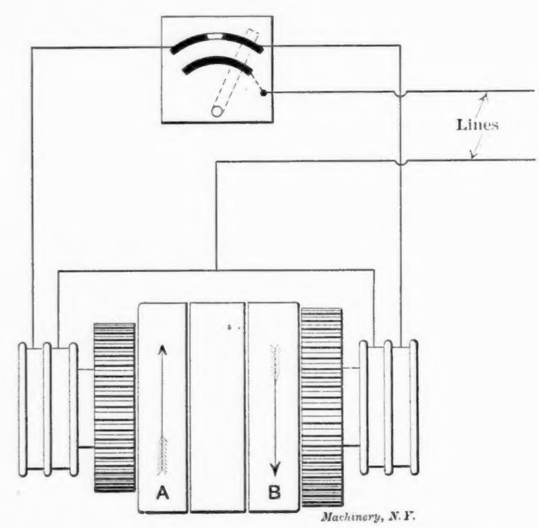


Fig. 2. Diagram of Magnetic Clutch and Connections.

I tried another way of overcoming the bad effects of residual magnetism and, so far as I know, with good result. A switch was used for reversing which would reverse the magnetism in one clutch for such a length of time as was necessary to kill

the residual magnetism. The current was then turned on in the other clutch and, though it is true that it took just as long to bring this clutch up to the point of saturation, it is also true that the clutch did not need to come to this point before it pulled the armature over to its side. Of course, this was because it did not have to overcome the residual magnetism on the other side. The actions of the switch were controlled by the dogs. The clutches used in that case had cork plugs projecting about 1/32 inch above the metal. This 1/32 inch lasted about 5 minutes, after which there was metal-to-metal contact. But this caused no trouble; in the first place residual magnetism was killed, and in the second place these corks acted as so many lubricators, thus reducing the wear to such an extent that it was not noticeable during the six weeks these planers worked under my eyes. I never heard of any trouble on that score with these same planers, though I could not swear that there isn't. The scheme of preserving an air gap between clutch and armature seems a very simple one, and so it is, as far as construction is concerned, but it has a number of consequences which are not desirable. First, there is to be considered the amount of current consumed by the magnets. Not that it makes much difference in cost whether one or two amperes are used, but more current means more heating of the clutch and, in order to avoid this, the clutch has to be made larger, which in its turn means more momentum at the moment of reversal. Another objection to the wood blocks, and to the cork, too, for that matter, is that the number of square inches of section of the path of magnetism must be a certain minimum, in order to get the necessary pull, for this pull depends on the number of square inches, and the number of lines per square inch. The wood blocks take up a large amount of area which, of course, is lost to the magnetic circuit. This makes the clutch larger again. If all the area of the clutch outside of the coil were metal its width could be less, the average diameter of the coil could be greater, and the heating effect would be less.

In order to drive the planer, and especially in order to reverse, there must be a certain torque on the driving shaft. This means that there must be a certain magnetic pull and a certain lever arm to work on. Suppose the clutch must transmit 20 H. P. at 100 R. P. M., then it transmits 79,200-inch pounds per revolution. Suppose the mean diameter of the friction blocks is 14 inches, then the pull at the blocks is 79,200 divided by the circumference of a 14-inch circle, which is 1,800 pounds. Suppose, further, that the coefficient of friction is 10 per cent; then the magnetic pull is 18,000 pounds. It is plain that this pull might be less, if the mean diameter could be increased. For this reason I suggest the alteration shown in Fig. 1 in dotted lines. The friction blocks are carried entirely outside the magnetic circuit, and are supported by a light framework of bronze. This frame needs only sufficient section for mechanical strength. It can be made as wide as may be necessary to give the friction blocks ample area. Being at a large diameter, it allows the clutch body to be made smaller. It adds to the weight of the revolving parts, and that at the worst point, far from the center; but the total moment of inertia of the clutch need not be more than where the friction blocks are in the magnetic circuit. This circuit, being uninterrupted, becomes more compact and therefore the leakage is less. I have applied this to a large planer, but it is still too early to give results. I hope to be able to do so later.

Some time ago I indulged in some experiments, in order to find out if it were not possible to quicken the action of the magnetic clutch. Though these experiments were not carried out systematically, and the results were not put to practice, their outcome may be of interest to some of the readers of this article.

Fig. 3 shows a diagrammatic view of the experimental arrangement. *A* and *B* were two clutches, strapped back to back, so as to make one sliding member. *C* and *D* were two other clutches, mounted on the shaft, and the necessary collars were provided, to prevent *C* and *D* from moving endwise. *A* and *B* were allowed one-eighth inch endwise movement. This arrangement formed an ordinary clutch with the armature, with the exception that the armature itself was provided with coils. Clutches *A* and *B* received current all the time, and always in the same direction. Clutches *C* and *D* also received current all the time, but this current could be reversed. The reversing was done by means of double-pole double-throw switches, which were mounted against the wall. When both handles were up, the armature was up against *C*; when the handles were down, it was up against *D*. The polarity of the magnets is indicated by *S* and *N*; *S* indicating the south pole and *N* the north pole. As similar poles repel, and opposite poles attract each other, the reversal of the poles *C* and *D* must bring the armature from one side to the other. It did this in such a short time that it was impossible to estimate the amount. The snap of closing the switches was apparently simultaneous with the blow of the shifting armature. In order to bring the armature in the central position, only one switch was reversed, making both clutches repellent. There was, of course, no blow, and it was therefore somewhat difficult to make observations as to the time consumed to push the armature away from the clutch. Though I am not prepared to say how little time it took, this much, I know that it was not possible for me to notice any time at all. Of course the experiment was crude, and finer observations would very likely show somewhat different results. The wiring of clutches *A* and *B* was now cut out and the clutch treated like an ordinary magnetic clutch, that is, the current was sent to only one side at a time. There was metal-to-metal contact, as the

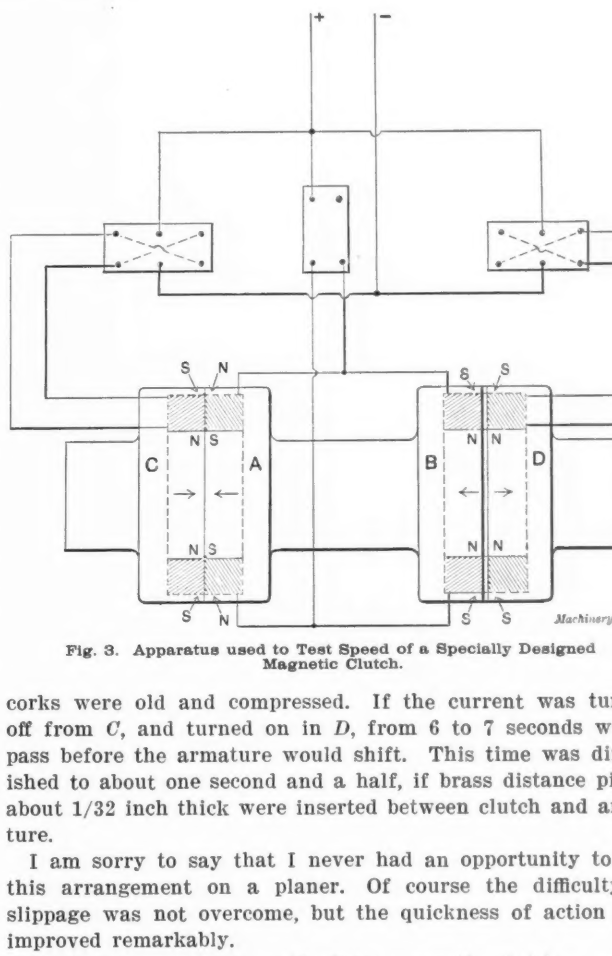


Fig. 3. Apparatus used to Test Speed of a Specially Designed Magnetic Clutch.

corks were old and compressed. If the current was turned off from *C*, and turned on in *D*, from 6 to 7 seconds would pass before the armature would shift. This time was diminished to about one second and a half, if brass distance pieces about 1/32 inch thick were inserted between clutch and armature.

I am sorry to say that I never had an opportunity to use this arrangement on a planer. Of course the difficulty of slippage was not overcome, but the quickness of action was improved remarkably.

Much more could be said about magnetic clutches, and I am sure very much could be done to improve them, but it is the question, whether the game is worth the candle, and whether life is long enough to spend a large portion of it on such an unthankful proposition.

One trouble experienced with magnetic clutches is their heating, when running on short stroke. It is nothing strange to see the smoke come out of the wood blocks. This is due to an insufficient area of the friction blocks. By placing the friction blocks outside the magnetic circuit, and by using the system of repulsion and attraction described above, most if not all of the troubles might be overcome. I hope later on to have a chance to try this, and to announce results, good or bad.

Pneumatic Clutches and other Devices for Planer Reversal.

The second kind of clutch is the pneumatic clutch. A complete description of this device would take up too much space, and is, in itself of so much importance that nothing but a complete essay will do it justice.

One of the most modern clutches, though old in principle is a Weston clutch, whose leaves are pressed against each other by compressed air. This construction has some inherent advantages. One is that the power transmitted can be made as large as desired by simply using more leaves. Another is that the leaves themselves offer a large frictional area, and that this again can be made still larger by simply multiplying the number of leaves. Care must be taken to have all parts self-contained on the driving shaft, so as not to bring any end pressure against the bearings, and another point of the greatest importance is to avoid leakage of air. These points, however, appeal to any good mechanic, and for that reason are not liable to cause such serious trouble as the magnetic clutch.

Another type of pneumatic clutch uses friction cones lined with wood blocks. Whatever clutch is used, the difficulties remain the same as with belt-driven planers, when high speed is the object. These difficulties might be overcome, if it were possible to devise a clutch of small bulk and weight, great starting power, and requiring only a small gear reduction. Thus far such a clutch has not been designed, and is not likely to be designed at any time, as some of its requirements are in direct opposition to some of the others. An attempt has been made to have the clutch nearly positive in its action, and thus give it great driving power, and then adding some device which will relieve the planer of shock. Various devices have been brought out from time to time, and were illustrated in different papers. They are used with both belt- and clutch-driven planers.

It has been proposed to make the table rack sliding under the table, and have it butt up at both ends against strong helical springs. I have never seen a planer thus arranged, but I fear that this arrangement is weak and liable to cause any amount of trouble. Others have placed the driving pinion loose on its shaft. Its hub was provided with a clutch tooth arranged like a cam, or rather, part of a helix. A similar clutch was mounted on the same shaft and made to slide over a key. A strong helical spring behind this clutch held the two cam surfaces in contact. When the pinion shaft began to turn, it had the tendency to take the pinion along, by the friction of the cam surfaces. It would thus start the table, but at only a part of the ultimate speed, as the clutch was sliding back at the same time. When this clutch was as far back as it would go, a projection would engage a corresponding projection on the pinion hub. From that moment on, the drive was positive. This arrangement again is weak and unfit for large and powerful planers. To show that I have given this subject some consideration, I will here describe a device which I got out some years ago, but though I consider it of interest, as showing one of the different roads pursued to get to a high-speed planer, I do not think it is the final solution. It is not patented and anybody is welcome to it.

Fig. 4 shows this arrangement. It consists of a cylinder, *A*, filled with heavy oil, in which a piston, *B*, moves. The piston is a nut moving over the screw, *C*. This screw is a part of, or attached to, one end of the pulley shaft of the planer, while the cylinder, *A*, is attached to the other end of this shaft. In this way, the device forms the coupling between the two parts of the shaft. The piston is guided by a key, *D*. It has a second keyway, diametrically opposite the first, which fits loosely over a throttling bar, *F-E*. This bar is made in two pieces, the stationary piece, *F*, and the rocking piece, *E*, and the latter is the real throttling bar. Suppose *B* is connected to the pulley end of the shaft. As soon as this begins to turn, the nut will have a tendency to go forward if it can do so; or, if not, then the nut will take cylinder *A* along, and thus drive the planer. In order to go forward, the nut must displace the oil in the cylinder and drive it from one end to the other. The oil has to travel through the opening left between piston and rocker bar *E*, which becomes smaller as the piston travels further, due to the slanting position of the rocker bar. This opening, being large at the beginning of the piston travel, but little resistance is offered to the movement of the piston, and of the cylinder, and thus, the planer remains at rest. As the opening becomes smaller the resistance increases, and soon a point will be reached where the piston starts the cylinder revolving. There is now a movement of both piston and cylinder, and the planer starts up, but not with its ultimate speed. As the resistance against the endwise movement of the piston increases, the speed of the planer increases also, until the piston is up against the other side of the cylinder, and the planer is traveling at its full speed. In order to give the piston a rapid endwise motion, the screw is made with a very steep lead (the angle of the thread was 30 degrees at the pitch diameter). The nut will travel in the opposite direction when

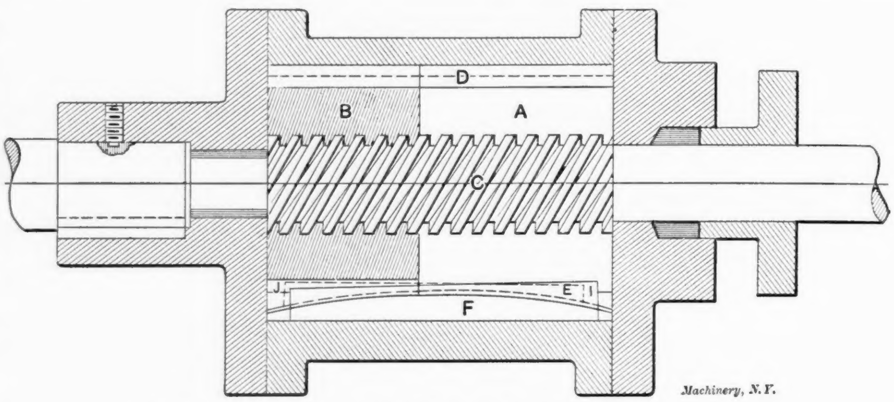


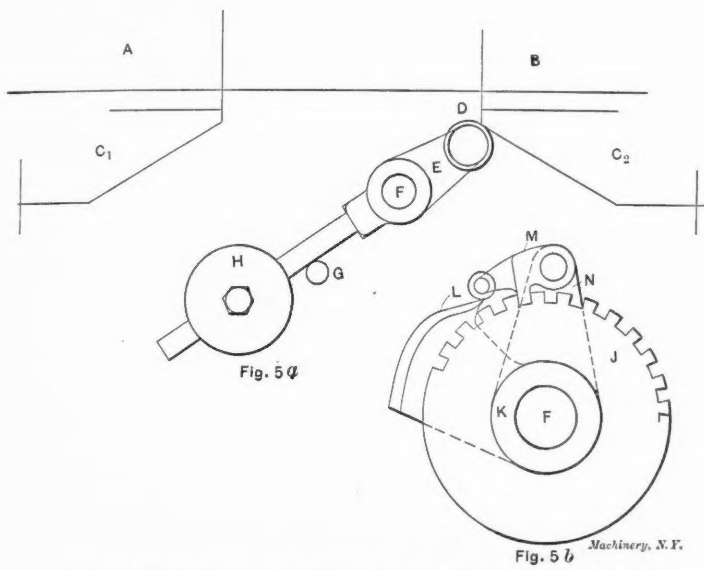
Fig. 4. Device to Provide for the Gradual Reversal of a Planer Table.

the planer reverses; it then forces the oil back to the other end of the cylinder. The taper of the throttling bar, however, is the wrong way. This is where the rocker bar does its fine work. The position of this bar at the beginning of the reverse strike is shown in dotted lines. As soon as the piston begins to move, pressure is brought to bear against the end, *I*, of the bar, while the end, *J*, is free to move. The bar will rock, therefore, and place itself in the proper position. This action takes place at the beginning of each stroke. The device worked very well, as far as I tried it, though I never tried it on a planer. Like a great many other devices of this nature, however, I consider it a source of weakness, and therefore to be avoided, as long as something better can be found.

Planers Driven by Direct-connected Motors.

The third group of planers are those driven by a direct-connected motor. For this style of drive, the motor is direct-connected to a driving shaft, without intervention of belts, clutches, or other mechanical devices. Such planer drives are now being built by the Electric Controller and Supply Co., Cleveland, Ohio. This system of driving a planer comes nearer to the ideal planer drive than any mentioned so far but it has some serious drawbacks. The motor is geared to one of the planer driving shafts. The planer is reversed by reversing the motor. Thus, all objections to starting up a planer at high speed, and to overcoming the momentum of revolving parts are obviated. When the planer is about to reverse, current is turned off, and the armature of the motor is short-circuited on a certain amount of resistance. Sometimes, not even this is done, and the current is turned on immediately in the opposite direction. Solenoid-operated switches control the motor; and they, in turn, are operated from a contact switch, operated by hand or by the dogs. The first solenoid-switch is operated direct from the tappet-switch; as soon as the motor has reached a certain speed a second solenoid-switch acts, cutting out starting resistance, etc.; the solenoids depending for their action on the speed obtained by the motor, i. e., on its counter-electromotive force. The field resistance is thrown in on the return stroke only, or on cutting and return stroke both, as may be desired. The drive is capable of working on short, as well as on long stroke; and it matters

not at how high a speed the table runs. This sounds very nice, and, in fact, it is nice, but there are certain draw-backs. Motors as they are commonly constructed have a speed variation not exceeding two to one. I do not mean to say that there are no motors with a greater speed range, but that they are the exception and not the rule. This means that with one of those motors the highest possible return speed is only twice as great as the lowest necessary cutting speed; for the speed range of the planer is, of necessity, the same as the speed range of the motor. Any motor with greater speed range must necessarily be larger and more expensive; and even then there is a limit. Four to one is an extreme range for one voltage. The Electric Controller and Supply Co. arrange some of their drives with the motor on the three-wire system, i. e., 110 and 220 volts, and thus increase the speed range; but this requires a large motor, the same as if the motor were run on one voltage only. Besides, the shops which use the three-wire system are a small minority. A planer driven this way was on exhibition at the St. Louis fair, and appeared to work very well—with the above limitations.



Figs. 5a and 5b, showing the Mechanism for Varying the Speed of Motor.

The idea of reversing the motor with the planer is not new. As early as 1899 I used a 20 H. P. reversible Westinghouse motor, direct-connected to a plate planer, for the Mare Island Navy Yard. This planer was built by the Niles Tool Works Co. The controller was furnished by the Cutler-Hammer Co., Milwaukee, Wis. The same style drive was used on a number of other plate planers, later on. Variable-speed motors were in their infancy then, so that it was not possible to use them for ordinary planers. (Plate planers plane both ways, and therefore run at *one* speed only.)

The Pond Machine Tool Co., one of the plants of the Niles-Bement-Pond Co., has in its shop a planer driven by a direct-connected Thompson-Ryan motor made by the Ridgway Dynamo and Engine Co., Ridgway, Pa. The motor has a speed range of 7½ to 1 on one voltage. It follows that the motor is a thing of might, and looms up in gigantic proportions alongside the planer. The tappet lever operates a controller, which starts a small motor. This motor, in its turn, operates the controller for the large motor. The entire outfit is large, expensive, and delicate. However, it does the work very nicely. It allows of a cutting speed as low as 20 feet per minute, and a return speed as high as 150 feet per minute. It may be called an interesting and successful experiment, from a mechanical standpoint. However, its cost and complexity do not recommend it for practical use.

A Solution of the Problem Proposed by the Author.

It has been my experience, that *one single* principle cannot govern a construction, if this construction shall serve a useful purpose. The science of engineering is the art of compromising. Some of this and some of that must be sacrificed, in order to get a really useful result. I am convinced that neither of the systems enumerated so far will lead up to a commercially successful planer. At the same time, all have certain points which it may be well to preserve. These considerations led me up to what I believe a good solution of the entire question. I combine the construction of the direct drive with that of the clutch- or belt-driven planers. Thus I get the advantages of the speed range inherent in the clutch-driven planer, with the speed range of the commercial motor. For example, take a planer driven by belts or a clutch, and having a cutting speed of 20 feet per minute, and a return speed of 60 feet per minute. These speeds are very moderate and obtainable in the largest size planers, without meeting any extraordinary difficulties. Let this planer be driven by a motor, running 500 R. P. M. Now, increase the speed of this motor to 1,000 R. P. M., and you have a planer with 40 feet cutting, and 120 feet return speed. Both the motor and the planer can be easily obtained. It remains to connect them in such a way as to get the greatest benefit out of the combination. Let us take a clutch-driven planer. While the planer is running on its cutting stroke of 20 feet per minute, I turn the field rheostat of the motor, until the planer speeds up to 30 feet. Just before the moment of reversing I turn the rheostat back to its original position, so that, at the end of its stroke, the planer runs again at 20 feet per minute. The planer reverses and starts up with an initial speed of 60 feet. Immediately after starting, I turn the rheostat to its highest point, bringing the motor up to its highest speed of 1,000 R. P. M., and making the speed of the planer 120 feet. Before the end of the stroke the speed is reduced again to 60 feet, and the planer reverses again at its low speed, thus doing away with the problem of braking and starting a large momentum. It is truly remarkable to see how quickly the planer responds to the rheostat, and how few inches of table travel are needed to slow down and to get speed up again. The slowing down is so rapidly accomplished that some precaution is necessary so as not to subject the planer to shock. The reason of this lies in the motor. When our motor runs 1,000 R. P. M., it does so in order to generate the necessary counter-electromotive force. The speed is high because the field is weak. The field strength becomes twice as great as soon as the rheostat is turned back to the first button. The motor, running now 1,000 R. P. M. in a strong field, generates a counter-electromotive force in excess of the voltage of the line, and therefore sends current back to the generator, and helps to drive other machines or light up the shop. This acts like a very powerful brake, and brings the motor down to its lower speed, almost instantaneously. The speeding up does not go quite so quickly, though, as I said before, only a few inches of table travel are required to bring the planer up to full speed.

The speeding up and slowing down of the motor at the proper time, can be accomplished in a great many ways, either electrical or mechanical. I have worked out various schemes, of which I will give one as an example. Fig. 5-*a* shows a sketch of the mechanism required. *A* and *B* are the dogs. Each one is provided with a cam-shaped projection, marked C_1 and C_2. These projections are in different planes, so as not to interfere with each other's action and each projection operates a roller *D*. These rollers are held in levers *E*, pivoted on the shaft *F*. The levers are counterweighted, and in state of rest, but against a stop pin, *G*. A spring may be used instead of the weight *H*. In fact, this would be preferable, as the weight ought to be adjustable, and as this might take up too much room. Fig. 5-*b* shows part of the mechanism removed, so as to show the rest more clearly. *N* is a lever, fastened to *E*, and carrying a pawl, *M*. This pawl works both ways (forward and backward) on the ratchet wheel *J*. There are, of course, two pawls and two ratchet wheels; one for each lever, *E*. Both ratchet wheels are keyed to the same sleeve, *K*. This sleeve revolves around the stationary shaft or stud *F*. A cam or shield *L* (one in front of one ratchet wheel and one behind the other) is adjustable on the shaft. It will be seen that the shapes of the cam and pawl are such that the cam throws the pawl out of engagement with the ratchet wheel. The projection *C* gives the roller, and thus the pawl, a certain fixed amount of motion at the end of the planer stroke; while the counterweight, or spring, brings it back again to normal position. The sleeve *K* carries a gear wheel or sprocket, not shown in the sketch, which operates the rheo-

stat. The style of connection depends on the shape of rheostat, the arrangement of the planer, and other local conditions. Calling the similar rollers corresponding to projections C_1 and C_2, D_1 and D_2 respectively; the two pawls, M_1 and M_2; ratchet wheels, J_1 and J_2, and shields L_1 and L_2, where in the sketch the letters D, M, J and L have been used to refer generally to these parts, the action of the device is as follows: Projection C_1 strikes roller D_1 just before the end of the cutting stroke and depresses it. This moves pawl M_1, and thus ratchet wheel J_1. The rheostat is brought to the off position. As soon as the table reverses, the lever is free to resume its position, and does so under the pull of the spring or the weight. The ratchet wheel is moved back, until the pawl is compelled to climb to shield L_1, when the ratchet wheel is left free. The rheostat is thus set to a point somewhere between "off" and "full on." Any speed of the motor, between minimum and maximum can thus be had by adjusting the position of the shield. It should be remarked here, that the shield L_1 when as far back as possible, still intercepts the pawl, so that the ratchet wheel is free at the end of the movement, regardless of the speed of the motor. At the end of the return stroke, a similar cycle of operations is gone through. This time it is projection C_2, which operates on roller D_2 and thus on ratchet wheel J_2. This ratchet wheel is free to move, as the other pawl is held out of engagement by shield L_1.

It will thus be seen that any speed of the motor, between its minimum and maximum, can be had at the beginning of either stroke, and that the speed of the motor is always brought back to its minimum at the beginning of the stroke. In this way, the planer can have any cutting speed ranging from 20 to 40 feet per minute, and that, in combination with any return speed ranging from 60 feet to 120 feet. This range might be increased on small planers from 20 to 160 feet, and, by taking a motor with a speed variation of 2½ to 1 (which is well within practical limits), a range of 20 to 200 may be obtained. It will further be seen that any cutting speed may be had in combination with any return speed, so that all conditions of work can be accommodated.

An advantage incidental to this arrangement is worthy of notice. As the feed takes place at the beginning of the stroke, and as the table speed is always the same at the beginning of the stroke, the feed will always be accomplished in the same number of inches of table travel. With an ordinary variable speed planer, the feed is either too fast at the high speed, or takes up too much of the travel at the low speed.

Though the arrangement, as described, is not applicable except with motor drive, I think it a great step forward toward the goal, viz., a practical high-speed planer. A similar arrangement might be constructed, operating on a variable speed countershaft, or speed box, instead of the motor.

* * *

VARIABLE SPEED MECHANISMS.—2.

CONE, DISK AND SPHERE DEVICES.

The following review of United States patents on variable speed mechanisms is not exhaustive in that it covers all such devices that have been patented; neither is it claimed that all the best devices are shown, but those devices are shown which it was possible to find in a comparatively limited time for the search of the patent records for the last fifty years. Owing to the peculiar, not to say defective, classification of United States patents it would be impossible to present a complete review of variable speed mechanisms without a complete search of nearly the whole field of patent art, which was impracticable. In the following review we have, therefore, confined ourselves mostly to those patents which in effect are designated as variable speed mechanisms as such, hence such devices which are incorporated as incidental features of other inventions have been generally ignored.

One of the first United States patents, if not the first, on variable speed mechanisms is that of J. Kello, April 1, 1815. He was granted a patent for "machinery to produce uniform motion by varying and irregular power." A few other scattered devices of this nature are the subject of patent record, and in 1855 J. T. Heacock patented a power metal drilling machine having a variable speed feed device consisting of cone pulleys in conjunction with a screw by which differential motion of the nut and the shaft was obtained. In this device, Fig. 11, power is transmitted from cone pulley G to the drill spindle through the spur gears H and E. The opposite end of the step cone pulley shaft is belted to the cone S, which in turn is belted to the cone P. This cone contains a nut mounted

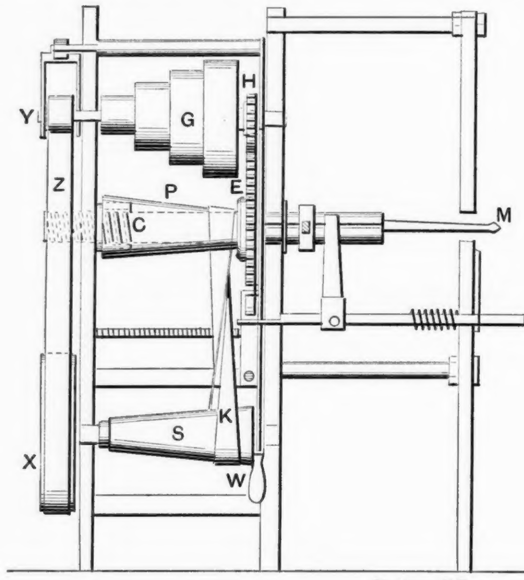


Fig. 11. Variable Speed Feed Device No. 13,845, by J. T Heacock, 1855.

upon the threaded end of the drill spindle. The drill spindle and the cone pulley turn in the same direction, and of course if their angular motion is equal, there will be no movement of the nut upon the screw. By varying the position of the belt K upon the two cone pulleys, a differential action was introduced by which the drill spindle was given a feed proportionate to the difference in motion of the pulley P and the spindle. It will be noted that the belt connecting the cone pulleys is twisted, which tends to equalize the stresses and prevent climbing.

J. A. Stoddard's device for changing speed, patented May 24, 1859, No. 24,159, covered a broad range in the claim. In his patent he claimed, as new (sic) means for graduating or varying speed by means of pulleys, or their equivalents, operated in connection with surface wheels or their equivalents in such

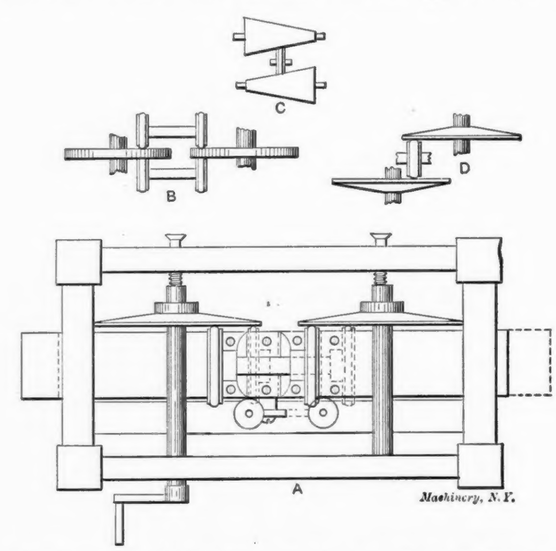


Fig. 12. Patent No. 24,159 of J. A. Stoddard, for Changing Speed, 1859.

a manner as to receive and transmit the motion at variable distances from their centers. It will be observed from the drawings, group Fig. 12, that his invention is not the simple friction and disk wheel, or one cone and friction wheel, but a combination of two such elements, whereby the speed-varying capacity is practically doubled.

In group Fig. 13 we have the well-known Sellers' patent, variable speed feed-device for lathes and other machine tools.

This was patented by Coleman Sellers, September 10, 1861, No. 33,283, and assigned to William Sellers & Co. In his patent specification, Mr. Sellers outlined the object of his invention as being to improve the transmission of motion by frictional contact, so as first, to insure the duration of the surfaces in contact, and to compensate for the inequalities arising from wear or inaccuracy of workmanship without affecting the position of the centers of rotation; second, to furnish a ready

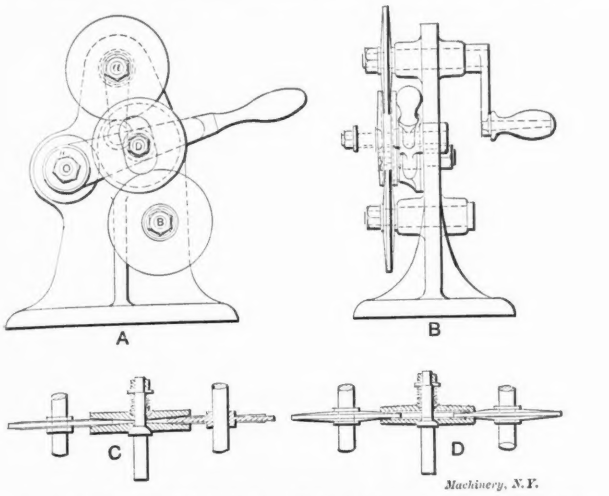


Fig. 13. Variable Speed Feed Device, Patented by Coleman Sellers, 1861, No. 33,283.

means of changing the velocity of the driver and driven; and third, to prevent the strain on the journals being increased much if any, over that due to the power transmitted, as in ordinary cog gearing. He criticised a belt as being objectionable for feed devices on lathes and other machine tools because of the necessarily short lengths between centers. The belt either had to be stretched so that its elasticity was destroyed, or an idler pulley had to be introduced so as to take up the slack; moreover, a short belt produced heavy pressure on the journals, and the third object of his invention was to

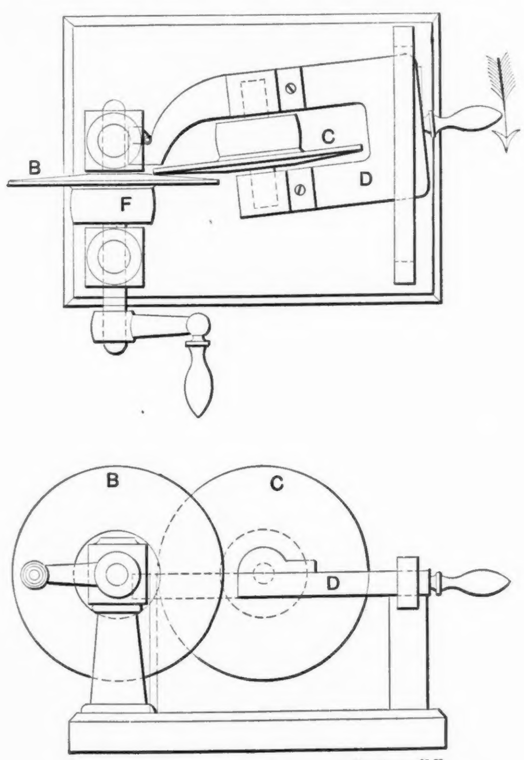


Fig. 14. Patent No. 128,773, granted to L. Wright, 1872.

overcome this. The second object, that of furnishing a ready means of changing the relative velocity of the driver and driven, could be obtained, as he pointed out, by means of the well-known device of two conical pulleys with a belt so arranged as to shift over their various diameters, but he had found this arrangement to be very objectionable in practice, owing to the tendency of the belt to climb up to the highest part of the pulleys, and because of the unequal stretching of the two sides of the belt. He also alluded to plain friction wheels and grooved friction wheel gearing, but said that these are not applicable where a ready change of velocity is desirable. In referring to the friction gearing of the disk and friction wheel type, having the shafts at right angles, he said that this was open to the objection of the plain friction gearing of having limited surfaces in contact, and also that surfaces were in contact which were running at different velocities. The essential features of the Sellers device are two

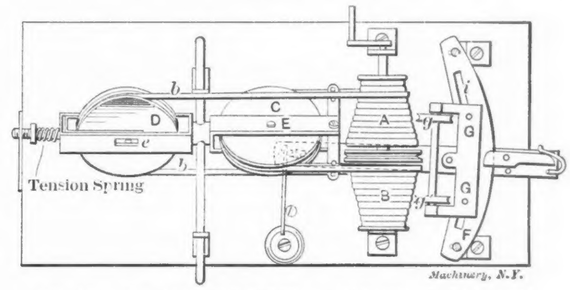


Fig. 15. W. H. Wilson's Patent, No. 140,751, issued in 1873.

disk wheels at a fixed center distance, connected by intermediate clamping disks, whose position can be varied at will. When the clamping disks are in the middle position, the velocity ratio of the driver to the driven is as 1 to 1, and the ratio varies proportionately as the clamping disks are shifted toward either the driver or the driven. In some respects the Sellers device is perhaps one of the best modes of transmitting variable motion, especially for light powers. When the disks are properly shaped, the grinding action is reduced to a mini-

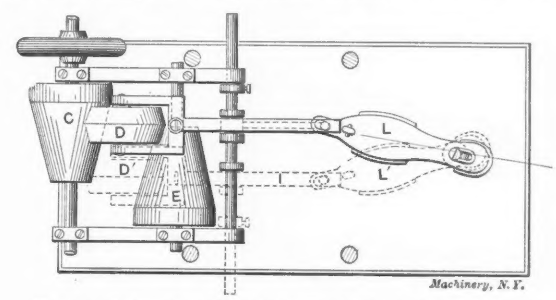


Fig. 16. Patent No. 142,504, issued to L. H. Olmsted, 1873.

mum. Unfortunately, however, it is limited in power carrying capacity, hence is not considered an acceptable device for transmitting large powers.

Patent No. 128,773, granted to L. Wright, July 9, 1872, utilizes two wheels having spherical surfaces which may be brought in contact at various radial distances by swinging one wheel and its frame upon a pivot. In the position shown in the upper view of Fig. 14, the velocity ratio of the driver to the driven wheel is at the maximum. By swinging frame *D*

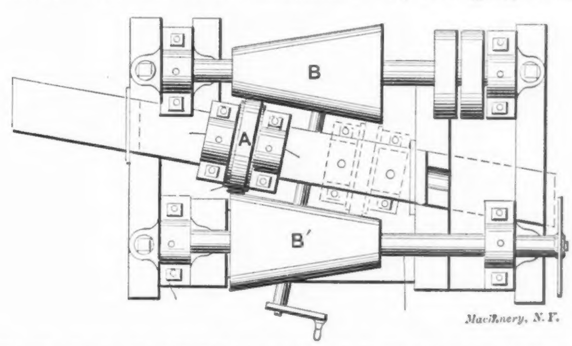


Fig. 17. Patent No. 148,066, granted to J. D. Husbands, Jr., 1874, for Improvement in Gearing of Stone Sawing Machines.

in the direction indicated by the arrow, a larger diameter of wheel *B* is brought into contact with a smaller diameter of wheel *C*, hence the velocity ratio is increased proportionately to the radial distances of the surfaces in contact.

In patent No. 140,751, granted to W. H. Wilson, July 8, 1873, motion is transmitted from one stepped cone pulley *A* to another *B*, having a coincident axis by means of a belt *b* running over idler pulleys *D* and *C*, Fig. 15. The inventor also claimed the alternative plan of transmitting motion from one cone to

the other by means of the friction wheels g and g', mounted in a frame G, which was arranged so that the two friction wheels could be brought in contact with the various steps of the cone from the largest to the smallest.

The device shown in Fig. 16 represents the invention of L. H. Olmsted, who was granted a patent, No. 142,504, Sept. 2, 1873. It consists essentially of two cone pulleys C and E, and a double-cone friction wheel D, with provision for varying the position of the friction wheel upon the cone pulleys by means of the foot piece L. It is obvious that the construction shown must be productive of much loss of power, inasmuch as the friction wheel is tapered to correspond to the taper of the cone, but the taper is in the opposite direction, hence all surfaces in contact, save those in the medial plane, are running at different velocities. This, it may be remarked, is the one great failing of almost all friction contact devices, and is largely the cause of their want of durability. The mere matter of transmitting power by the friction wheel is not so severe as the grinding action that is inevitable when surfaces are in contact which do not or cannot travel at the same rate.

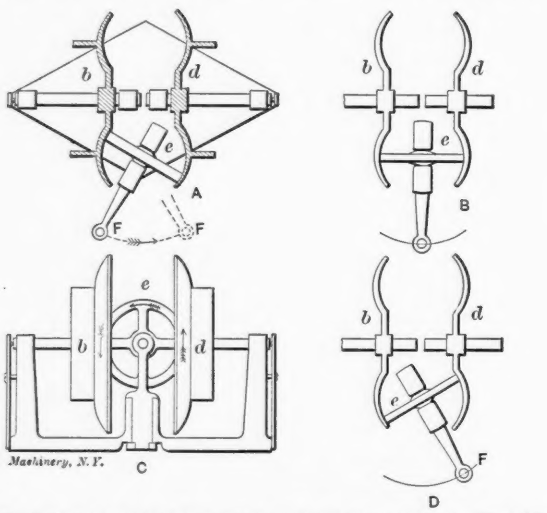


Fig. 18. Patent No. 197,472, granted to C. W. Hunt, 1877, for Variable Speed Countershaft.

In Fig. 17 the patent No. 148,066, granted to J. D. Husbands, Jr., March 3, 1874, for improvement in the gearing of stone-sawing machines, is of a more practical nature. In this the friction wheel A is made cylindrical in shape, hence the difference in movement of surfaces in contact with the cones, is not as great proportionally as in the previous device, just described.

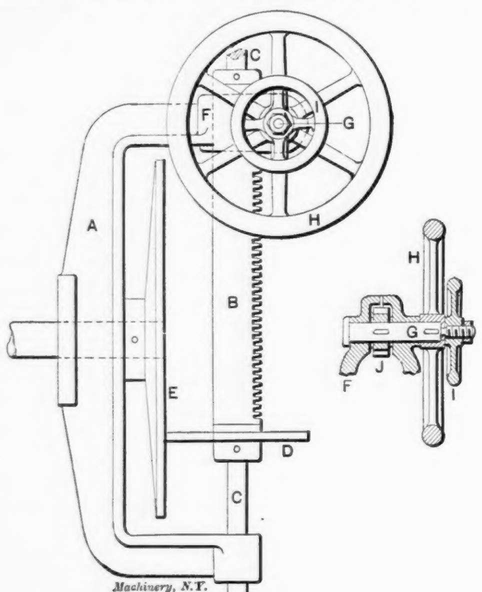


Fig. 19. Patent No. 258,533, granted A. Gordon and T. Reiss, 1882.

On November 27, 1877, C. W. Hunt was granted patent No. 197,472 for the variable speed countershaft, group Fig. 18. It consists of two disks b and d, which are made with grooves, a cross section of which is the arc of a circle. These disks embrace between them within the torus thus formed the friction wheel e. The angular position of this wheel determines the velocity ratio between the driving and driven disks. This idea has been the subject of numerous later patents.

In Fig. 19 we have the well-known variable feed device, largely used on the machine tools built by the Niles Tool Works, for which patent No. 258,533 was granted to A. Gordon and G. T. Reiss, May 3, 1882. The construction is obvious from the drawing. The principal element of novelty is the means provided for shifting the friction wheel D across the face of the disk wheel E. The hand wheel H, carrying the pinion J, meshes in the tubular rack B, and this provides means for controlling the position of D. Small wheel I is for locking the hand wheel pinion in any desired position.

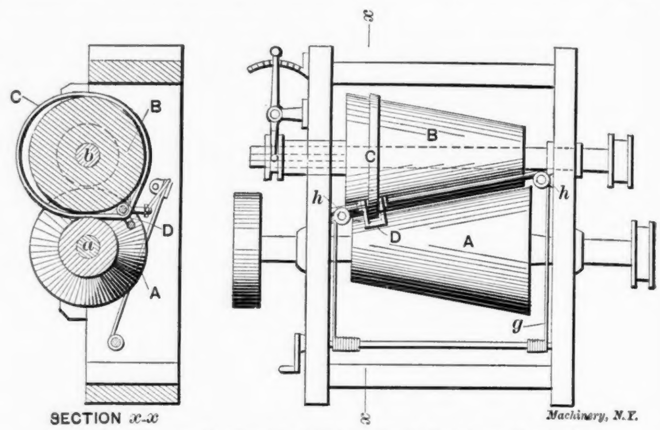


Fig. 20. Device Patented in 1884 by W. E. Laird, as No. 299,231.

In Fig. 20 patent No. 299,231, granted to W. E. Laird, May 27, 1884, we have two opposing cone pulleys, A and B, mounted closely together, but actual contact is made through the belt or band C, hence the longitudinal position of this band determines the surface of cones A and B, which are in virtual contact, and thus the velocity ratio is changed by shifting C from one end to the other of the cone B.

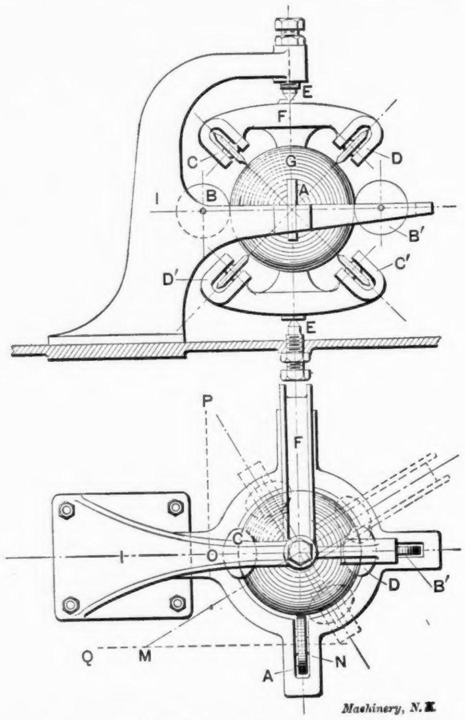


Fig. 21. Patent No. 312,171, H. S. Hele Shaw, 1885.

The device shown in Fig. 21 was granted to H. S. Hele Shaw, Feb. 10, 1885, No. 312,171, and is designated as an "apparatus, whereby the relative motion of two or more bodies may be varied in any required manner, independently of their actual motion." This interesting device patented by Mr. Shaw, is described at length in his specification, which is illustrated with twelve drawings. Fig. 21 represents the more general form of the arrangement of the mechanisms. It consists of a sphere G held in position by friction wheels C, C', D, and D'. A friction wheel A is supposed to be the driver and B the

driven. The position of the frame F determines the actual velocity ratio of the driver and driven wheels. In the positions shown by the full lines it is obvious that the friction wheel B would receive virtually no motion whatever from the motion of sphere G when transmitted by A. By shifting the position of the frame F to that indicated by the dotted line, more and more motion is given to B, depending upon the angle of displacement, and as the frame is shifted to an angle of 90 degrees from that it formerly occupied, it is claimed that the conditions existing in the first place are completely reversed, or in other words wheel B would have an infinite motion to that of A.

* * *

AUTOMATIC TWIST DRILL GRINDING MACHINE.*

The machine described in the article here abstracted is manufactured by Frederic Schmaltz, of Paris. It is shown in half-tone in Fig. 1.

It is well known that it is quite impossible to obtain the best results from twist drills which have been ground by hand, as any deviation from the correctness of center, or difference between the cutting angles of the lips of the drill, ruins the truth of the hole and imposes considerable stress on the drilling machine. To get the best results out of twist drills two chief conditions must be fulfilled, supposing the quality and temper of the steel to be the best. These conditions are, first, that the two cutting edges simultaneously come into contact with the work throughout their whole length. This condition is only fulfilled when the two cutting edges are elements of the same cone. In standard practice the included angle of this cone is 118 degrees. Second, the lip clearance or the backing off of the metal back of the cutting edges should vary with some regularity from the point of the drill where it is greatest to the periphery where it is least acute. The cutting angle of the drill is determined by the amount of this clearance; it varies from 50 to 60 degrees, 59 degrees being the cutting angle most generally employed, although recent experiments with high-speed steel drills would indicate that a somewhat smaller angle would be more advantageous. This has not been adopted into standard practice, however, and where a drill is to be employed for various metals, the angle had best approach 60 degrees, according to our French contemporary.

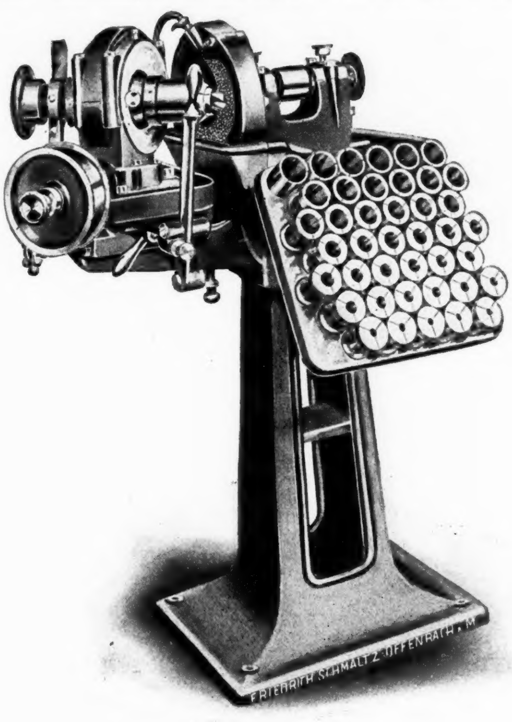
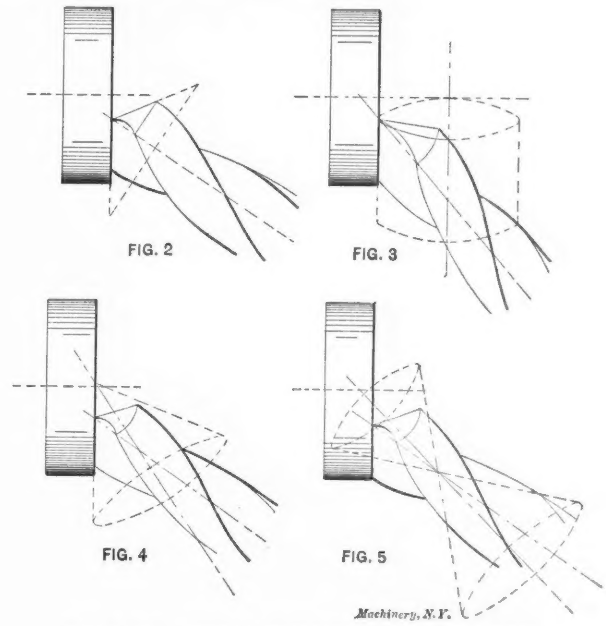


Fig. 1. Schmaltz Twist Drill Grinder.

Four possible methods of grinding the cutting edges of a twist drill are represented diagrammatically in Figs. 2 to 5. Fig. 2 represents a drill being ground on the plane surface of a grinding wheel, the drill being turned about its own axis, thus giving to the extremity of the drill the form of a cone, as shown. This construction allows no clearance back of the cutting edges determined by the intersection of the conical surface with the spiral flutes of the drill, and the drill will be terminated by a sharp point, which would not cut. In Fig. 3 the drill is not only turned about its own axis, but at the same time a circular movement around an axis parallel to the face of the grinding wheel is impressed upon it, this motion producing a certain clearance back of the lips. This clearance



Figs. 2 to 5. Possible Methods of Grinding Twist Drills.

will, however, be the same from the point of the drill to the periphery, and as according to all authorities the clearance should vary from the point to the outside circumference, this will not do. In Fig. 4, which is a construction commonly used in American grinding machines, the drill is given a conical movement around an axis somewhat oblique to the axis of the drill, as shown. The apex of the cone is above the point of the drill, and the cutting edges of the drill are elements of this cone. In this construction a clearance is generated, which increases uniformly from the periphery to the point of the drill.

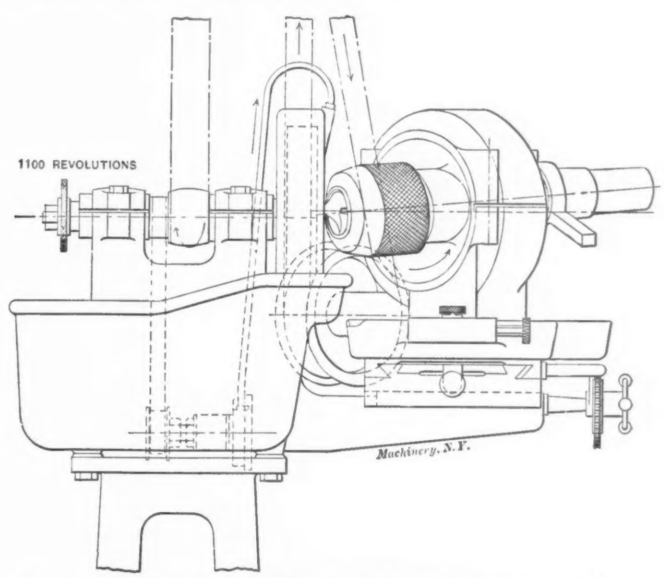


Fig. 6. Side Elevation of the Schmaltz Grinder Broken Away, showing Arrangement of Belting.

In Fig. 5, which is the construction used in the machine about to be illustrated, the cone described has its apex behind the point of the drill. This construction apparently gives very much the same sort of variable clearance as that given by the construction in Fig. 4. The variation in clearance does not seem to be as great from the point to the periphery, but, on

* Portefeuille Economique des Machines, March, 1905.

the other hand, the clearance of a drill should not be excessive at the point or it will cut too rank. According to the *Portefeuille Economique*, the grinding with this inverted cone construction is done by the wheel cutting the cone close to its apex, giving a pronounced clearance with the least angle of opening of the cone.

After this introduction, it may be said that the automatic machine in question, as shown in detail in Figs. 6 to 11, carries a grinding wheel mounted on a belt-driven spindle which

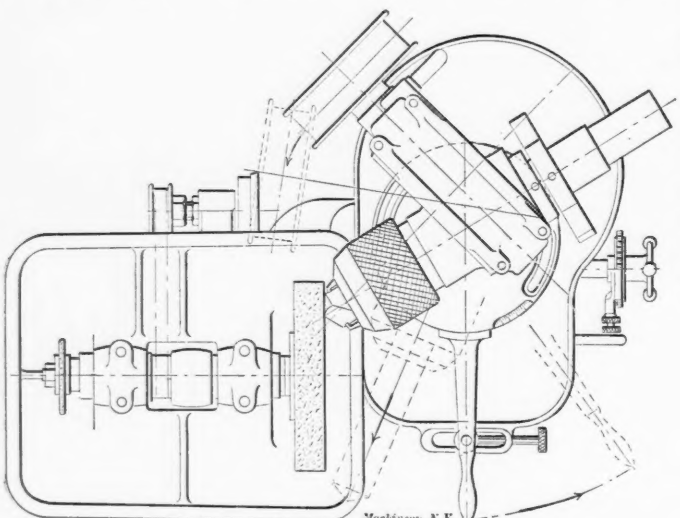
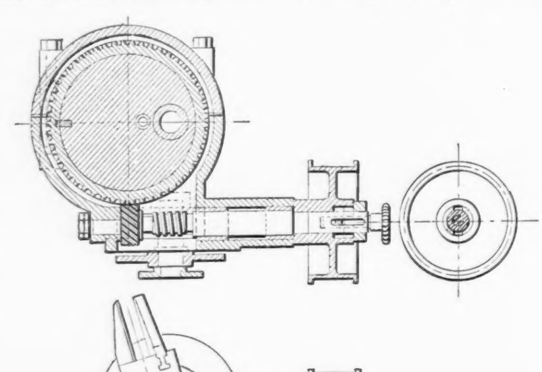


Fig. 7. Plan View of the Grinder.

is given 1,100 revolutions per minute. A belt-driven pump is provided, for cooling the wheel and drill. The drill to be ground is mounted upon a carriage, which also is operated by an independent belt from the countershaft.

The drill is given four movements, of which three are simultaneous and continuous, and the fourth is produced periodically at each half turn. These movements are, first, a continuous conical revolution as described above, in which the drill to be ground is brought against the face of the grinding wheel; second, a periodical rotation of the drill about its own axis. This is so arranged that the drill turns about its own axis as many times in the course of one complete rotation about the axis of the cone as there are cutting edges to be ground, so that each lip is successively brought in contact

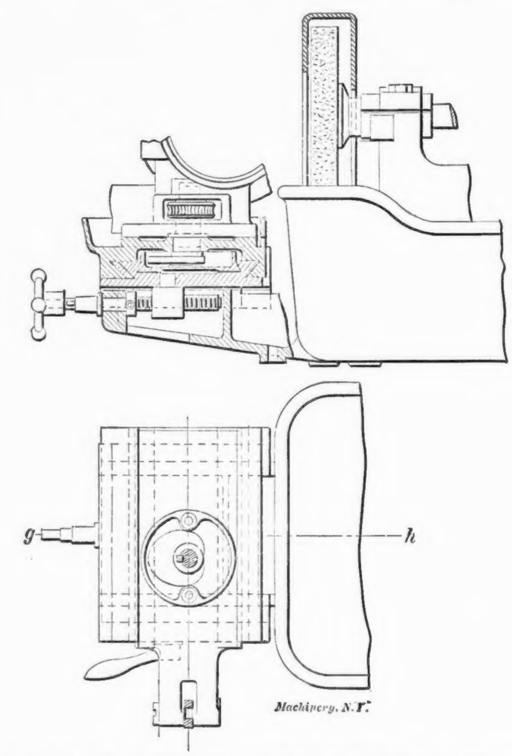


Figs. 8 and 9. Fig. 8, Longitudinal Section on line a—b, Fig. 9. Fig. 9, Plan and Horizontal Section through Drill-carrying Socket.

with the face of the grinding wheel. Third, a back and forth movement of the carriage carrying the drill, parallel to the face of the grinding wheel, so that the grinding is equally distributed over the whole face of the wheel, and the latter is thus equally worn. Fourth, a feed motion perpendicular to the face of the grinding wheel.

All these movements are derived from the belt-driven pulley shown on the carriage. The spindle carrying this pulley turns a disk, as shown in Fig. 8, by spiral gearing. In this disk is placed the socket carrying the drill to be ground, obliquely to the axis of the disk. As the disk revolves, a cone is described like that indicated in Fig. 5. This is the first movement mentioned above. For the second, the drill-carrying socket is provided with a lever, back of the disk, which lever encounters, each revolution, a finger fixed in the cap of the disk. The socket is thus obliged to turn about its own axis. This lever is double, as shown in Figs. 7 and 9, for a twist drill having two edges, triple for a three-lip drill, etc., so that the cutting edges of a drill are all equally ground, each revolution of the disk.

For the cross motion of the carrriage a horizontal worm-wheel meshes with a worm carried on the pulley spindle, as shown in Figs. 8 and 10. A cam fixed to the arbor of the worm wheel produces the cross motion of the carriage by its action on oppositely disposed rollers carried by the cross slide as shown in Figs. 10 and 11. The fourth, or feed, movement, is obtained automatically through the agency of a pawl and ratchet, actuated by the cross movement of the carriage. The drill may be fed into the wheel by hand, through the hand



Figs. 10 and 11. Fig. 11, Plan View of Carriage, and Fig. 10, Elevation and Section on Line g—h, Fig. 11.

wheel shown in Figs. 6 and 10, after the automatic feed has been thrown out by lifting the catch. To replace the drill after grinding by a new one, the carriage is revolved about a pivot by means of the hand lever shown in Fig. 7. When the drill comes into the position shown in dotted lines, it may be easily removed.

The drill was held in place on the first machine built of this type by a sort of chucking device, in which, at first, three holding lugs were used. To obtain a better centering of the point, this was replaced successively by chucks carrying four and then six lugs, a better centering being obtained by each. It was finally determined that the most precise centering was to be obtained by means of split bushings, bored to the diameter of the drill. The machine is at present furnished with a series of such bushings, having diameters varying by millimeters. These bushings are shown in Fig. 1, carried in a stand which is arranged for them on the side of the machine. In operation the proper bushing is forced into place round the drill, when it is easily tightened about the drill by the knurled nut shown. By this method drills are quickly and exactly centered, and easily replaced.

In grinding, the drill is presented to the grinding wheel at the angle previously determined, usually 59 degrees. During each rotation the drill turns about its axis twice through an angle of 180 degrees, if a double twist drill, or three times

through 120 degrees if it has three cutting edges, and successively presents each of these cutting edges to be ground. By this automatic arrangement, the lips are certain to be of exactly the same length. The grinding wheel is covered by a casing to prevent the spattering of water. Part of this casing is hinged, and may be lifted in case it is desired to grind a piece of work on the edge of the wheel. A small thin grinding wheel having rounded edges is carried on the rear of the grinding wheel spindle for the purpose of reducing the thickness of the web of a drill between the cutting edges, when this has become too great. This is common to most drill grinding machines and is necessary because the flutes of a drill become more shallow as they approach the shank to increase the strength. Hence, after a drill has been ground a number of times, the web between the cutting edges becomes too thick.

This apparatus is a wet grinder, an automatic pump taking water from the basin formed by the top of the structure, and throwing it on the lips of the drill being ground. The machine is constructed in four sizes.

* * *

VARIABLE SPEED MOTORS.—12.

THE ELECTRO-DYNAMIC COMPANY'S INTER-POLE VARIBLE SPEED MOTOR.

WM. BAXTER, JR.

The variable speed motor made by the Electro-Dynamic Company, of Bayonne, N. J., is constructed so as to permit wide speed variation, through field regulation, without producing sparking at the commutator brushes. As has been explained in several of the articles of this series, the range of variation obtainable by means of field regulation, in motors of the ordinary design, is limited practically to a ratio of about two to one, owing to the fact that a greater variation causes the brushes to spark to an injurious extent. In the Electro-Dynamic inter pole motor double this range of variation is obtained without producing any visible sparking. The result is accomplished by means of the addition of small poles that are placed intermediate between the regular poles of the motor. The construction can be understood at once from an inspection of the photographic illustration, Fig. 1, which shows the motor field complete, the armature being removed so as to

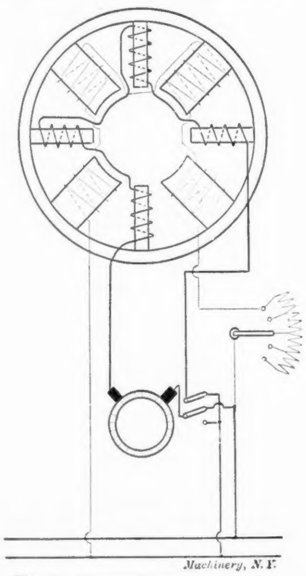


Fig. 2. Circuit Connections of the Motor.

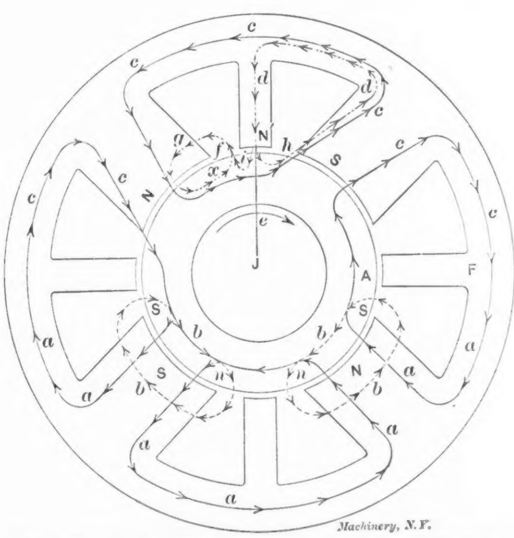


Fig. 3. Diagram showing the Path of the Magnetic Fluxes.

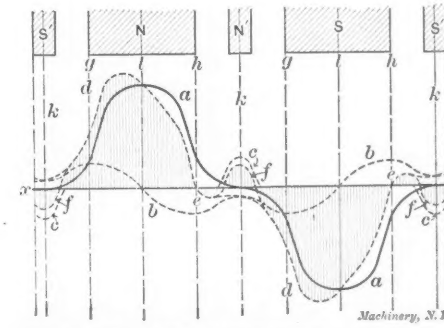


Fig. 4. Curves of Magnetism showing Action of the Inter Poles.

more clearly reveal the position of the inter poles. The coils wound upon these inter poles are connected in series with the armature and as a consequence their magnetizing effect varies with the strength of the armature current. The circuit connections of the motor are clearly shown in the diagram, Fig. 2, in which the outline of the field is shown at the top and the armature is represented by the small circle directly below it.

The Office of the Inter Poles.

The office of the inter poles is to provide a commutating magnetic field that will increase and decrease in accord with the armature current, and thus prevent sparking at the brushes, and it is for the purpose of attaining this end that the coils are connected in series with the armature.

The first impression an electrical engineer would gather upon inspecting this motor would be that it operates upon the same principle as the Thompson-Ryan motor, made by the Ridgway Dynamo & Engine Company, but such a conclusion is not strictly correct. Both constructions accomplish the

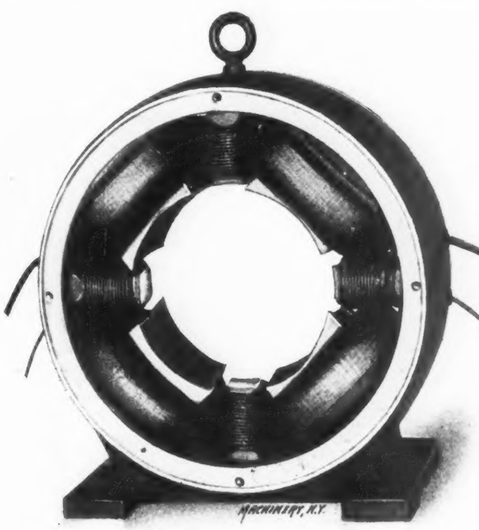


Fig. 1. Motor Field, showing the Inter Poles.

same result, but the action in the two is different. In the Thompson-Ryan motor, the balancing coils act to neutralize the armature re-action, and at the same time to develop a commutating field of the proper magnitude to obtain sparkless commutation. In the Electro-Dynamic Company's motor the inter poles act to develop a commutating magnetic field of the proper magnitude to produce sparkless commutation, but do not counteract the distorting effect of armature re-action upon the field magnetism. The distortion of the latter magnetism by the re-action of the armature does not result in any injurious effect upon the action of the motor; that is, it does not reduce its capacity, or its efficiency, hence, in so far as practical results are concerned, the action of the two systems is the same.

Principle of Action of the Inter Poles.

The principle upon which the inter poles act in the Electro-Dynamic motor can be made clear by the aid of Figs. 3 and 4. In these diagrams we have shown the paths of the several magnetic fluxes when acting alone and also when acting in combination with each other. Fig. 3 shows in outline the field *F* and the armature *A* of the motor. If we assume that the armature is held stationary, and that current is passed through the field coils, there being no current traversing the armature, the magnetic flux developed by the field coils will flow in the paths indicated by lines *a a a a* in the lower half of the diagram. If we shut off the current from the field coils and pass a current through the armature, then this current flowing through the armature coils will develop magnetic fluxes that will flow in the paths

indicated by lines *b b b*. If currents are passed through the field coils and the armature at the same time, the flux developed by the field coils will not be located along the lines *a a a* and that developed by the armature will not be along the lines *b b b b*. Magnetic fluxes will not flow in opposite directions through a magnetic circuit, and they have a decided objection to crossing each other's paths, hence, when the field and armature coils are both traversed by currents, the former

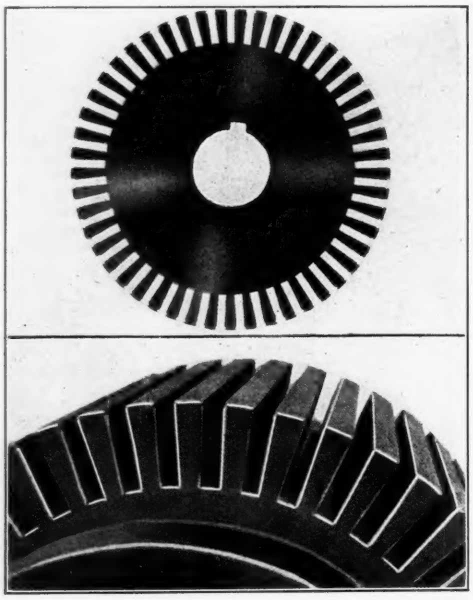
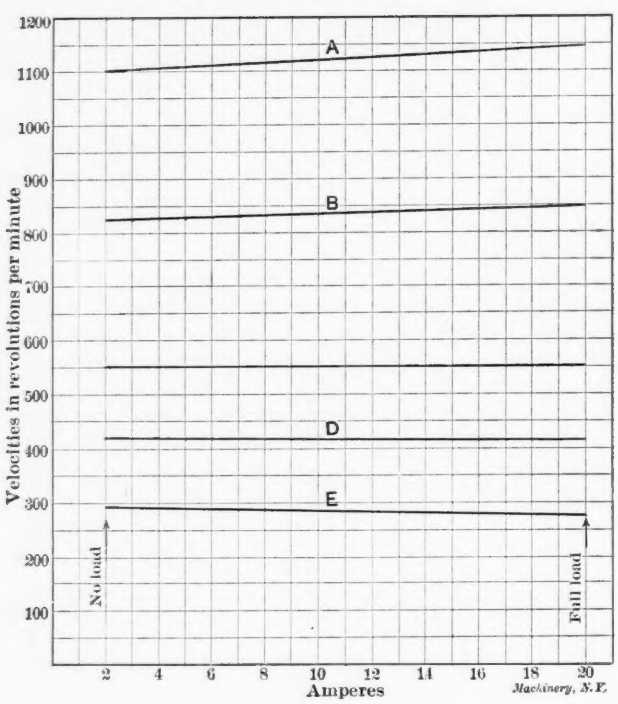


Fig. 5. Portion of Armature Core, showing Deep Slots to hold correspondingly Large Coils.

try to push the latter out of the way, and the latter try to treat the former in the same manner. If the armature and field are traversed by currents flowing in the proper direction to cause the rotation to be clockwise, as indicated by arrow *e*, the re-action of the two magnetic fluxes will result in shifting the field magnetism back to the position indicated by the lines *c c c* in the upper half of the diagram. The greater part



Speed curves of variable speed inter-pole motor at different velocities.
Fig. 6.

of the armature flux will follow this path, although some of it will circulate in the shorter path indicated by curve *g*. The stronger the armature current the greater the displacement of the field flux, so that while a very weak armature current might not cause the lines *c* to deviate much from the position of lines *a a a*, with a sufficiently strong current they could be shifted even further than shown in the diagram.

The Effect of the Armature Flux.

Although the effect of the armature flux is to shift the field flux toward the position of lines *c c c*, all the magnetism is not forced to follow this path; as a matter of fact the magnetic flux will issue from the whole polar surface, all the way around to the edge *f*, unless the armature re-action is enormous. When a motor is running under practical conditions, the displacement of the flux is not sufficient to drive it away from edge *f*, in fact some of the flux will pass to the armature even beyond this point, forming what is called the commutating fringe. In motors of the common type, the commutator brushes are set so as to short circuit the armature coils as they pass a point slightly in advance of *f*, for example on the line *x*, and the strength of the magnetic field at this point is sufficient to perform the operation of commutating the current effectually, that is, for a given strength of armature current. If the armature current increases, the field flux is driven further in the direction of lines *c c c* and the flux at line *x* becomes weaker, when it should be stronger to effect perfect commutation. If the armature current is reduced the result will be just the opposite, the flux at *x* will be increased, when it should be reduced.

Path of the Inter-pole Flux and its Commutating Action.

In the Electro-Dynamic motor, the inter pole *N'* develops a flux that follows the path *d*, a very small portion of it probably takes the path *i* from the end of the pole pasing around path *g* to join *c*. This flux being developed by the same cur-

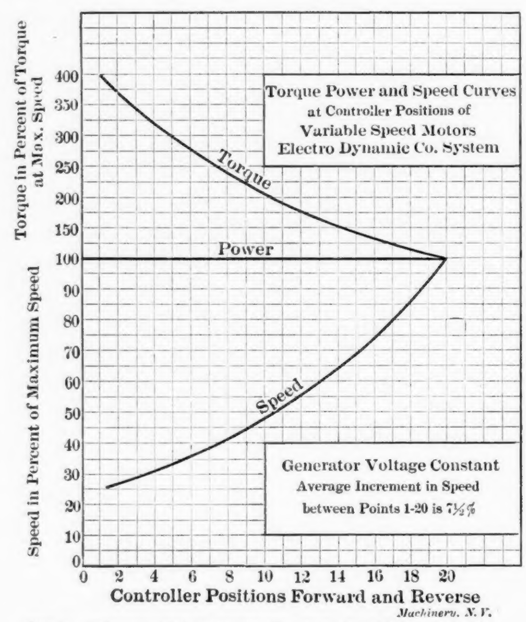


Fig. 7. Torque and Velocity Curves for Different Loads.

rent that passes through the armature increases and decreases in accord with this current and as a result it is the proper kind of a flux to perform the act of commutation. If the commutator brushes are set so that the short circuited armature coils are directly under the center of *N'*, the commutation will be perfect for all strengths of armature current because the flux of *N'* will be weak when the armature current is weak, and strong when the armature current is strong. Without going into a complete dessertation on the subject of commutation it is not possible to make perfectly clear the reasons why the magnetic flux of the inter pole *N'* can produce perfect commutation on account of its varying as the armature current varies; but as this subject has been discussed in previous articles of this series it need not be treated in this connection any further than to say that to prevent sparking it is necessary that the current flowing in the commutated coil be reduced to zero, and be replaced by a current of like magnitude flowing in the opposite direction, while the coil is short-circuited. If the armature current is weak, the act of commutation will consist in stopping a weak current and building up another weak current flowing in the opposite direction, the whole action being accomplished while the commutated coil is short circuited. If the armature current is strong, a strong current must be stopped and an equally strong one must be generated in the commutated coil while this coil is short cir-

cuited. The commutating magnetic flux accomplishes this result by inducing in the commutated coil an electro-motive force, that is opposite in direction to that of the current flowing in the coil at the instant when it is short-circuited, and great enough to stop the current and build up a reverse current to the same strength by the time when the short circuit of the coil is broken. The stronger the armature current, the greater the electro-motive force required to be induced in the commutated coil to reverse the current. The magnetic flux of pole N' increases and decreases with the armature cur-

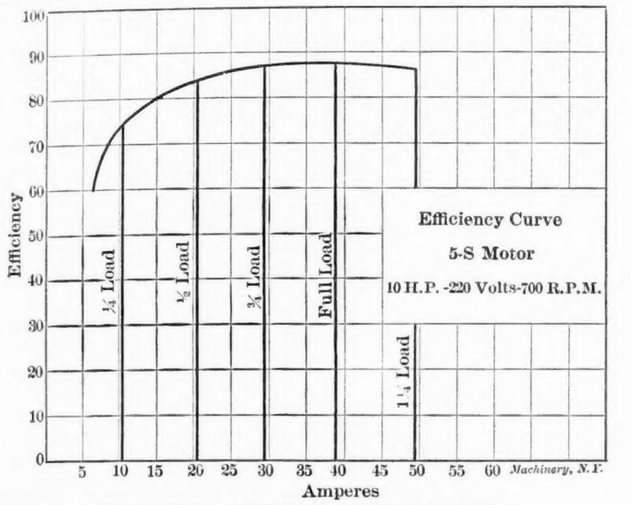


Fig. 8. Curve of Efficiencies for Various Loads.

rent, hence, it will induce in the commutated coil an electro-motive force of the proper magnitude for any strength of armature current, assuming of course that N' is properly proportioned.

As is stated in the foregoing if a motor of the ordinary design has the brushes set so as to short circuit the armature coils when they pass line x, the flux at this point will be too weak for perfect commutation when the armature current is strong, and too strong when the armature current is weak. This is the case with a constant speed motor, but if the armature is made with a small amount of wire on it, so that its reaction upon the field is small, the imperfection in the commutating action will not be sufficient to produce noticeable sparking with any variation in the strength of the armature current that is likely in practice. If, however, the motor is

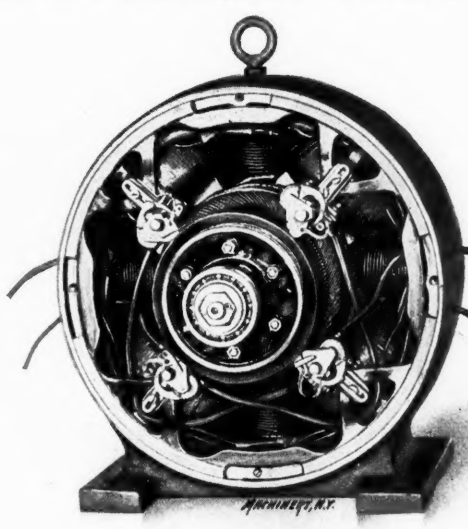


Fig. 9. Motor with Front Frame Removed, showing Ball Bearings.

of the variable speed type, then the case will be decidedly different because the variations in velocity are obtained by weakening the field magnetism, and this weakening is not slight, but on the contrary it is great; thus to double the speed the field magnetism is reduced to one half. Now if the field is reduced to one half, it is equivalent to doubling the strength of the armature magnetization so that to obtain a good running motor of the ordinary type with a speed variation of two to one it is necessary to make the armature decidedly weak, and the field correspondingly strong. With the construction of Fig. 3 it can be seen that it makes no difference to what extent the field magnetism is reduced, within reasonable limits, because the act of commutation is performed by the magnetism of the inter pole N' and this is developed by the armature current, hence it will be in proportion to the latter without regard to what the relations between the field magnetism *a a a* and the armature magnetism *b b b* may be.

Inter Pole Action Unaffected by Armature Reaction.

That the variation in the strength of the field or the armature magnetism, or both has no effect upon the action of the inter pole N' can be made clear by means of Fig. 4. In this diagram, the shaded outlines at the top represent the poles of the motor, the large ones being the main poles and the small ones the inter poles. Curve *a a* represents the field magnetism when there is no current flowing through the armature coils. Curve *b b* represents the armature magnetism when there is no current passing through the field coils. Curves *c* represent the magnetism of the inter poles. Curve *d d* represents the actual magnetism of the motor when running, with current in the armature and field, and the curves *f f* represent the commutating field produced by the action of the inter poles.

The field magnetism curve *a a* is shown high under the poles because at this point it is strong owing to the fact

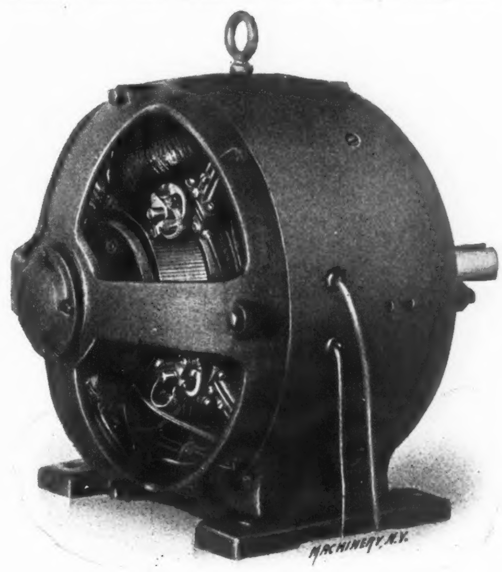


Fig. 10. External Appearance of the Inter-pole Motor.

that the greater part of the flux follows this path on account of the greatly reduced magnetic reluctance. As a matter of fact, in a well-designed motor, the strength of the magnetic field between the poles is much less than is indicated by curve *a a* and under the poles it is more uniform, so that to be strictly correct, the curve under the poles should be flatter, and in the interpolar spaces it should be lower; but the shape shown will serve to illustrate the actions better.

The armature magnetism is shown as stronger under the edges of the poles than at any other points because such it really is, owing to the fact that all the flux that passes beyond the poles and cuts through the interpolar space has to traverse a long air path, as can easily be seen by noting the length of air path of the lines *b b* in Fig. 3.

The curve *d d* is obtained by simply adding curves *a a* and *b b*. As will be noticed, this curve, which represents the actual magnetization of the motor, passes to the opposite side of the line *x x* just beyond the front edge of the main poles. This is due to the distorting effect of the armature magnetization. If the inter poles were not provided the brushes, if placed ahead of *h*, that is, in the position of line *x* in Fig. 3, would not work well at all, because the commutating action would be just the reverse of what is required; that is, the current in the commutated coils instead of being reversed, would be kept flowing in the same direction and would be increased in strength. To obtain proper commutation with a field dis-

torted as much as shown in Fig. 4, the line x would have to be shifted back of the edge h of the pole.

The inter-pole magnetism c will produce a commutating flux f of practically the same magnitude for any amount of armature re-action when the motor is working within its practical range, owing to the fact that the strength of the armature as well as the field magnetism at the center of the interpolar space, on line k, is very much lower than that under the poles of the machine, hence the changes in it cannot be very great. The reverse magnetization e is produced by the armature flux that circulates in the path g of Fig. 3, together with the small portion i of the inter pole magnetism.

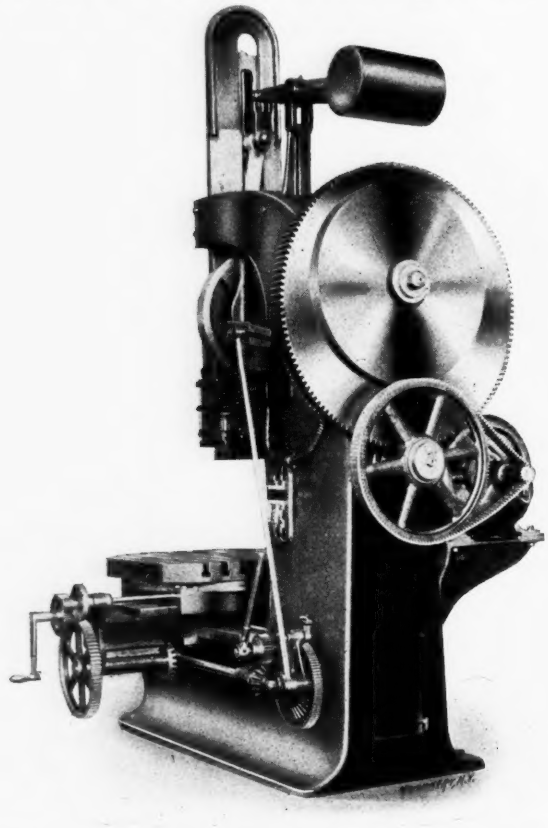


Fig. 11. Twenty-one-inch Bement-Miles Slotter driven by four H. P. Inter-pole Motor; Speed Ratio, 3 to 1.

Owing to the fact that the armature re-action, unless unreasonably great, does not affect the action of the inter poles, the amount of wire wound upon the armature can be much greater than in motors of the ordinary type and in this way the capacity of a machine of a given size can be considerably increased. That advantage is taken of this fact is shown in Fig. 5, which is a photographic illustration of a portion of the armature core of the motor.

Speed Regulating Effect of the Inter Poles.

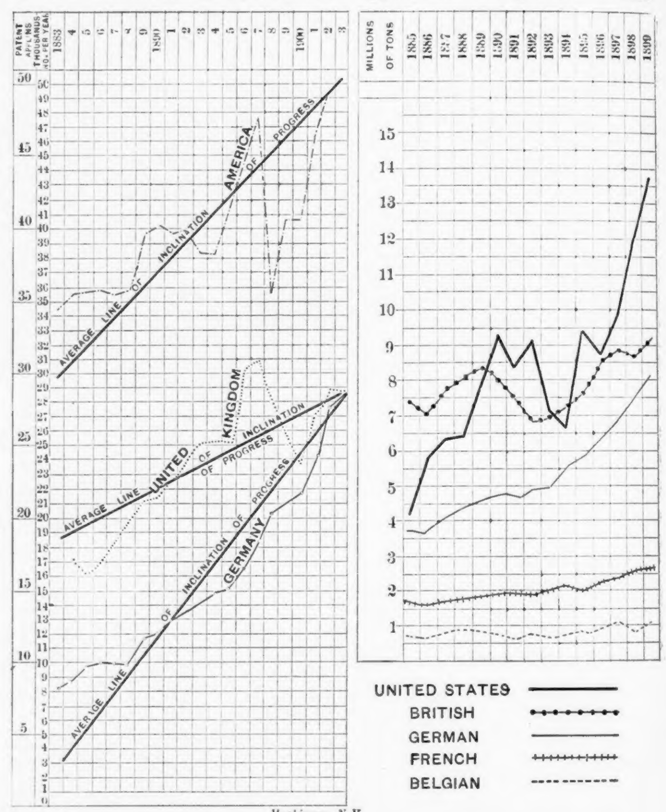
In Fig. 3 it will be noticed that the flux of the inter pole passes out of the armature at h, flowing in the same direction as the field flux, hence it helps the latter to develop the c.e.m.f. The flux entering the armature from N' can be made to neutralize itself, by setting the brushes so that one-half of it cuts through the armature in front of the commutated coil and the other half back of the coil. By shifting the position of the brush, the flux passing from the inter pole into the armature can be made to add to the effect at h or to deduct from it; therefore, by properly setting the brush the assistance given to the field flux in developing the c.e.m.f. can be varied as desired. The result of this is that the motor can be adjusted so that at any given velocity, the speed will remain constant for all loads. For higher velocities, the speed will rise as the load increases, owing to the fact that the e.m.f. developed by the inter-pole flux will increase with the speed; and for lower velocities the speed will drop with increasing load, but not as much as it would if the help of the inter poles were removed. The inter-pole motors are adjusted so that the speed remains constant for all loads when the velocity is about half way between the highest and the lowest. The actual relations for different loads and velocities are shown in the curves A to E of Fig. 6.

The relations between torque and velocity for different loads are shown in Fig. 7 and the efficiencies for all loads within practical limits are given in Fig. 8. The motor is made to run in ball bearings which are clearly shown in Fig. 9, the front frame that holds the bearing in position being removed. Owing to this construction not only is the friction reduced, but the dimension of the motor in the direction of the shaft is considerably reduced. The external appearance of the motor is shown in Fig. 10; it is compact, and small for its capacity, as can be judged from Fig. 11, which shows a 4 H. P. motor with a speed variation of three to one, mounted upon a 21-inch Bement Miles slotter.

* * *

INVENTION IN GREAT BRITAIN, GERMANY AND AMERICA.

In a discussion in the British trade papers concerning the lack of encouragement of the British inventor, as compared with his American and German rivals, the proposition was made by Mr. B. H. Thwaite that the true index of any nation's industrial and commercial position was not that of the axiom of Disraeli—the state of the chemical industry—but the condition of the iron and steel industry. Mr. Thwaite thought there was also another index almost as valuable, namely, the degree of activity of a nation's inventive faculty, represented by the numerical proportion of applicants for patents. The



Charts showing Comparative Inventive Activity and Production of Pig Iron.

charts here given, prepared by Mr. Thwaite, show the progress of inventive activity in America, the United Kingdom, and Germany during the twenty years ending 1903, and the comparative progress of the five great iron producing countries in the production of pig iron from 1885 to 1899. From the former it will be seen that the rate of increase of inventive activity of Germany has been slightly more rapid than that of the United States, with Great Britain left some distance in the rear. The attitude of the British patent office is one to discourage many inventors. For instance, under recent legislation the British inventor is compelled to sub-divide his claims so as to cause him to apply for a number of patents in place of one. Taking out patents is a very expensive matter in Great Britain, and it is claimed by many that the cost under the new act referred to will be greatly increased. Moreover, the granting of a British patent does not even now insure validity. The German inventor, on the other hand, is stimulated to enterprise by the German Mercantile Banking system, by which selected inventions of promise are developed and commercially introduced under the best conditions to secure success.

SHOP TRAINING FOR MECHANICAL ENGINEERING.

DESCRIPTION OF THE COURSE IN SHOP WORK AT THE WORCESTER POLYTECHNIC INSTITUTE.

HOWARD P. FAIRFIELD.

The great interest manifested in the shop education of a student in engineering gives evidence of the need for such a training. That this should be a broad rather than a special training seems to be the opinion of all employers, and of men of experience in this line of engineering. This is so generally the fact that the few who hold the opposite view are hardly in evidence. The majority of technical colleges or schools

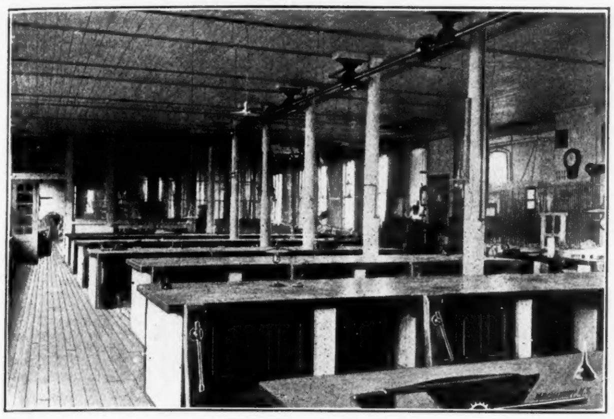


Fig. 1. Work Benches in the Pattern Shop.

recognize this need and provide for it by developing some sort of a shop course. These shops, however, seldom reach the really effective point in instruction because of the necessary expense for equipment. The cost of purchasing new and up-to-date tools and machines makes it expensive to give instruction in modern methods, and on this account the shop course in most colleges of engineering, falls far below the degree of importance which it deserves.

The shop training at the Worcester Polytechnic Institute is in its methods noted for two things, (*a*) the broad and comprehensive grasp of the needs of an engineering graduate, and (*b*) the necessity for keeping up to date in shop management and methods. The shops are made so far as the student is concerned strictly educational. They are an integral part of the department of mechanical engineering, the professor of mechanical engineering being also director of the shops.

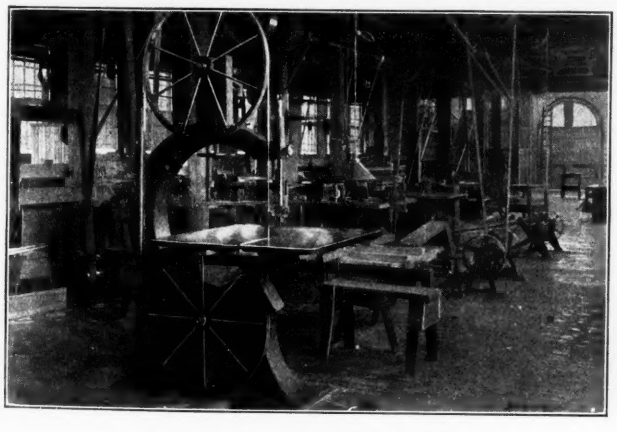


Fig. 2. Another View in the Pattern Shop.

The shops are also run as a manufacturing concern employing a number of machinists, molders, and pattern makers. To successfully and economically produce and market a line of machinery requires a modern shop and equipment. A modern system of cost accounting and methods of production must obtain, and all the various costs and operation from the patterns to the selling of the finished machines must make up a systematic whole. It is on this basis that the shops of the Worcester Polytechnic Institute are equipped and run both as a commercial enterprise and as a means to an educational end.

The students are assigned definite hours in these shops, depending entirely upon the engineering course they are taking; that is to say, whether they are fitting themselves for mechanical, electrical, or civil engineering. The students in mechanical engineering get the most extended shop course. The training of the students in the shops is in charge of instructors in mechanical engineering, and as such they are responsible to the head of the department. Four instructors are provided and their business is to teach and help the student in acquiring the broadest and most complete knowledge that the instructor can give during the time assigned to his special work. The instruction is at all times such as the future engineer will need and not that suited to the mere artisan or workman. All instruction in shop work, methods, and management is called "Shop Practice" and is given in the shops, the shop drafting room, and in the shop office. The whole equipment of the shops is at the service of the instructors, the offices as well as the work rooms, and are all and severally used as required. The needs of the student are studied, and as far as it is possible he is provided with the necessary training to fit him for his work as an engineer.

The complete shop course in mechanical engineering is about fourteen hundred hours, and of this amount one-half is at the disposal of the instructor in machine construction. Five hundred and forty hours of the total time is called "Summer Practice" and is given during the month of June, one hundred and eighty hours in each of the first three years. This summer practice is continuous work, ten hours per day until completed, and is common to all the institute courses. For students taking the course in civil engineering it means field work, and for those taking the course in chemistry, chemical laboratory.

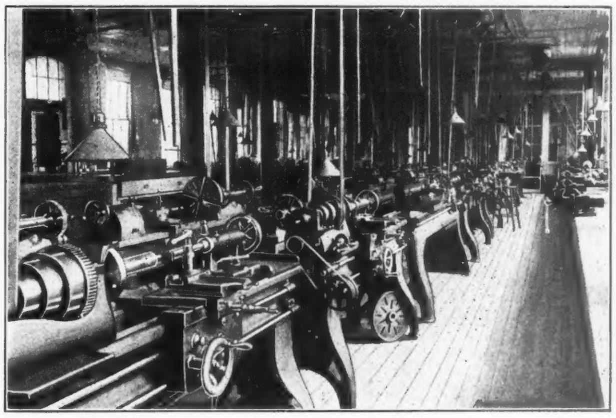


Fig. 3. The Lathe Department in the Machine Shop

The views here shown give some idea of the extent of the floor space provided in the shops and the means at hand for training the students. Figs. 1 and 2 are views in the pattern shop. All first-year students receive instruction in pattern making throughout the year, this work being common to all the courses.

Figs. 3-4-5 are views in the machine shop. Students in the mechanical, electrical, and civil engineering courses receive instruction in this department in varying amounts, according to the course they are pursuing.

Fig. 6 is a view of the forge room, and Figs. 7 and 9 are views in the foundry. The mechanical, electrical, and civil engineering students also receive instruction in these branches.

As already mentioned, the work in pattern making is common to all students since the selection of courses is not made until the end of the freshman year. During the sophomore year the electrical engineering students receive shop instruction during both terms and the sophomore summer practice. The civil engineering students receive shop instruction during a portion only of the sophomore year. For students in mechanical engineering the shop practice forms a part of each of the four years of the course.

The freshman, or first year work, is designed to train the student in the principles of pattern making, the work beginning with the simpler forms of patterns and advancing as rapidly as the ability of the student warrants.

While most colleges believe it is necessary to give the stu-

dent a course in elementary wood work or joinery before commencing his pattern making course, it is not thought advisable to do this at Worcester. If the graduate is to become a teacher of manual training he undoubtedly needs to go through an elementary course in woodwork, but no such training is needed by the future engineer, and thus no time is here given to this work.

Pattern making is first taken up in its relation to the foundry and the questions of draft, shrinkage and allowances for finish are studied. The question of selecting the proper lumber and the method of its use in pattern construction is also included in the elementary pattern making course.

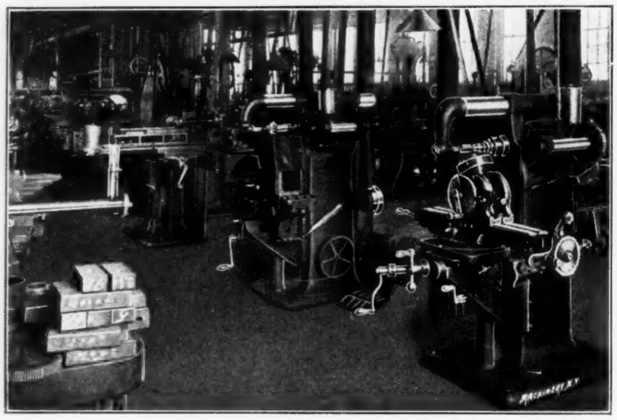


Fig. 4. Machine Shop. Instruction in Milling

The advanced pattern making work includes a study of cored sections and the pattern as a part of a machine. While at work on the above course the student is kept in close touch with the foundry and the machine shop and every point made by the instructor can be checked by actual commercial examples. In the pattern shop all those machines that are common to pattern making establishments are open to the student and pattern costs can thus be made a part of his course.

A short course in foundry work is introduced in the freshman year. This course is used principally as a method of checking his work in pattern making and is a beginning of a thorough course in foundry usages and management.

In all the freshman shop work the student is trained to think for himself and his standing in the classes is regulated to a large extent by his ability to do this. Throughout all the shop training the student is made to understand that as a

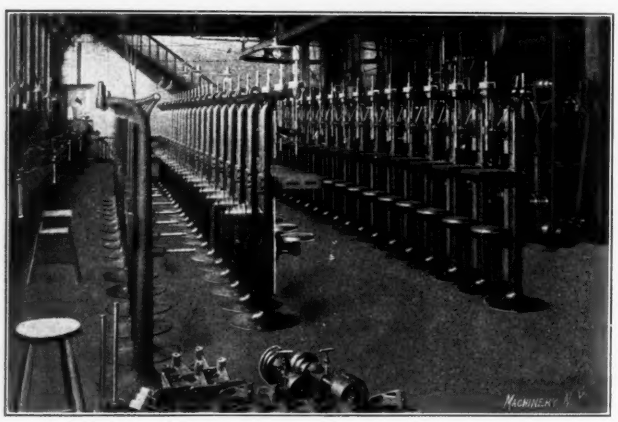


Fig. 5. Sensitive Drills, Completed and in Process of Assembling.

graduate engineer he must be able to think and plan not only for himself but for others as well; in fact that he is to become the leader rather than the mere workman.

As in the other branches the foundry is carried on as a business proposition and castings made for outside parties. This adds variety to the work and gives the student an opportunity to take part in the production of work which runs from light machine parts to large charcoal iron castings weighing over two tons.

At the beginning of the sophomore year a start is made in machine practice or metal work. The course as arranged is expected to give the student a knowledge of the use and capacity of all the ordinary machine tools. The students have access to and receive instruction in the use of lathes for both light and heavy work, planers, shapers, milling machines, gear cutters, boring mills, grinding machinery, drilling and chucking machinery and flat turret machines of various makes and sizes; also all the small tools, jigs and fixtures that are necessary for the economical manufacture of a line of light machinery. Instruction in the use of these machines is by lectures and by using the machines themselves. The student is expected to be able to do work on any machine in the shop by the end of his sophomore practice.

During the sophomore year the students start on their forge

Fig. 6. A View in the Forge Room.

work, which includes the effect of heat on different metals, welding wrought iron and various grades of steel, tempering and annealing, tool making, heat treatment of high-speed steels, work with power hammer and small drop forging.

In the second term the use of high-speed steels is taken up with the classes and the students provide themselves with tool holders for this purpose. A study of cutting speeds, feeds and their effect upon production, is begun at this time, and the student is shown three limiting features of production, (*a*) what the machine will stand, (*b*) what the work will stand, and (*c*) what the tool will stand.

A beginning is also made during this year in the practice of estimating the time necessary to perform work in the different machines used. Upon being assigned a job, the student receives the necessary blue prints and rough stock, and he is required to estimate the necessary time for completing the

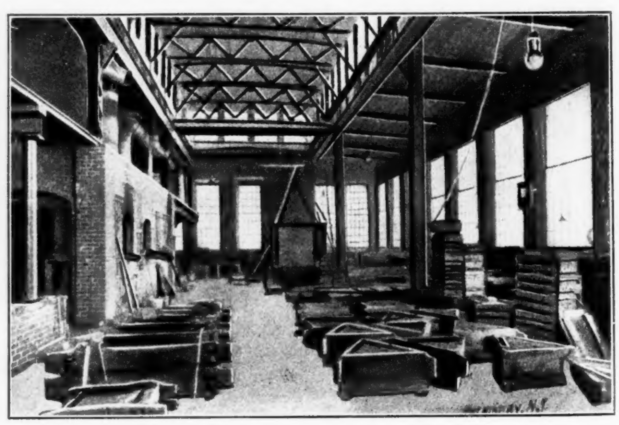


Fig. 7. A Part of the Foundry.

job. The instructor then gives him a card similar to that shown in Fig. 10. This card is retained by the student until the job is completed when it is handed in and stands as a record on the job for that student. On the back of the card the student fills in each day the kind of work done and the time taken, the total of these amounts finally appearing upon the face of the card together with the grade given his work. A complete index is kept and these job cards show at any time the student record, a different color being used for each year of the course. While the estimating of time must at the first be pure guess work it soon ceases to be so and the student is encouraged to keep notes of the speeds and feeds, used in

performing the several jobs, that he may more intelligently estimate the next one. This is in line with the idea that the student is later in his course to take up the study of estimating time of building parts of a machine.

In the foundry the sophomore also has at his command an opportunity to compare costs of molding, coremaking, and the many operations incident to foundry practice. The foundry is a new, modern, and well lighted building, and is equipped with two cupolas, a large core oven, a travelling crane, molding machines, core making machines, fans, rattlers, etc.

The work of the junior year is largely advanced machine construction, and a study is made of jigs and fixtures and their usefulness in duplicate work. The use of high-speed steels is given further consideration. Besides their regular line of work the shops build experimental machines for outside parties. This gives the student an opportunity to study the methods of doing work of this kind. He also learns of the increased cost made necessary when building a first or experimental machine over the cost of producing such machines in large numbers, when the jigs and tools have been developed, and the workmen have become skilled in their use.

Operation.	Details of Operation.	Method of Operation.	Machines Used.	Remarks on Setting Up of Machines.	Tools Used.	Speed in R. P. M.	Speed in feet per minute.	Feed per Rev.	Time for Setting Up Machine.	Time for Setting Up Work.	Time for Setting Tool.	Time for Taking Out Work.	Time for Detail of Operations.	Total Time.
Inspecting and Cleaning.	......	......	Emery wheel	......	......	....	.....	....	..	..	..	..	..	27″
Centering.	Prick-punching	Approximate center and test	Lathe (centers)	......	Scratch awl center sq. prick-punch	....	.....	....	..	..	..	..	2′ 20″	4′ 18″
	Center Drilling	Change work when drilled	Drill Press	......	$\frac{1}{8}$″ Drill	....	.....	....	1′	10″	..	5″	65′	
	Center Reaming	Change work when reamed	Drill Press	......	60° Reamer	....	.....	....	5″	10″	..	5″	22″	
Machining Surface A.	Roughing cut	Change tool after cut	14″ N.O.W Lathe	True up Live Center	Large Face Plt. Hook driver, half center H. S. R. Tool	283	100 max.	Hand Feed	5″	15″	20″	..	20″	1′ 36″
	Finishing cut	Change tool after cut	14″ N.O.W. Lathe	......	T. S. Side Tool	283	100 max.	$\frac{3}{128}$	..	..	20″	5″	10″	
Machining Surface B.	Roughing cut	Change work after cut	14″ N.O.W. Lathe	Remove half-center Line up centers	Full center R. H. offset H.S.R. Tool	283	100	$\frac{1}{64}$	5′	15″	15″	5″	1′ 5″	5′ 35′
	Dimension cut	Change work after cut	14″ N.O.W. Lathe	Line up centers	R. N. offset T. S. D. point	283	100	$\frac{1}{64}$	5′	15″	30″	5″	1′ 5″	
	Scraping cut (2 cuts)	Change work after cut	14″ N.O.W. Lathe	Line up centers	Lathe Scraping Tool	175	63	$\frac{1}{32}$	5′	15″	1′	5″	2′ 20″	
Machining Surface C.	Finishing cut	Change work after cut	14″ N.O.W. Lathe	......	R.H. offset H.S.R. Tool	175	63	Hand Feed	..	15″	20″	5″	2′	2′ 20″
Machining Surfaces D, E, F.	Roughing cut Surface D	Change tool after rough cut on Surface F	22″ Pond Lathe	......	H. S. R. Tool Special Jig	75 47	120 max.	Positive Feed $\frac{3}{64}$″	5′	15″	20″	..	5′	24′ 10′
	Roughing cut Surface E					47	120		..	..	10″	..	1′ 5″	
	Roughing cut Surface F					47	120		..	..	10″	..	20″	
	Finishing cut Surface D	Change tool after finish cut on Surface F			T. S. R. Tool	75 47	120 max.	Positive Feed $\frac{3}{64}$″	..	..	20″	..	5′	
	Finishing cut Surface E					47	120		..	..	10″	..	1′ 5″	
	Finishing cut Surface F					47	120		..	..	10″	..	20″	
	Final cut Surface D	Change work after cut			Scraping Tool	25	63 max.	Positive Feed $\frac{3}{64}$″	..	..	40″	5″	9′	
													Total time,	38′ 26″

Fig. 8. Table of Calculations made by Seniors in Estimating Cost of Production of a Machine Part. Totals are on the basis of 50 pieces.

In all this work the student is led from one method to another, one step at a time in a regularly advancing course. Each student has the same opportunities as his classmates but is allowed to advance as rapidly as his abilities allow.

The shop work for the first term of the senior year is devoted to a study of methods of cost reduction and cost accounting in the shops. (Fig. 8 illustrates the results of the study

of the cost of production as made upon a machine part by two seniors.) Fifty pieces are taken as delivered by the foundry and are machined complete, ready for the stock room. A comprehensive study is first made of all the machines, tools and fixtures that might be used in making the part, and the piece is then machined in several ways to determine which tools, machines, etc., are best suited to the work. After finishing these preliminary tests, and having settled upon the machines to be used, tests are made to determine the most desirable feed, speed, and depth of cut to use. To find this, several combinations are tried, each being carried to a point where the machine, the work, or the tool, will stand no more. The results of these tests are studied and a decision arrived at for the best and most economical speed, feed, and depth of cut for roughing out the several surfaces. Various methods of finishing the surfaces are then tried and the best one adopted. The order in which the operations should be performed, is next taken up and different arrangements tried and a conclusion drawn. The several conclusions are then discussed by the instructor and students, and when finally decided upon, are tabulated. The time taken to perform the various operations is carefully taken, notes are made of all that occurs in each case, and the results tabulated. Fig. 8 represents the results of the completed tests and the conclusions arrived at. Besides this a full written report illustrated by sketches of operations is also handed in to the instructor, explaining in detail the tabulated figures and reasons for conclusions.

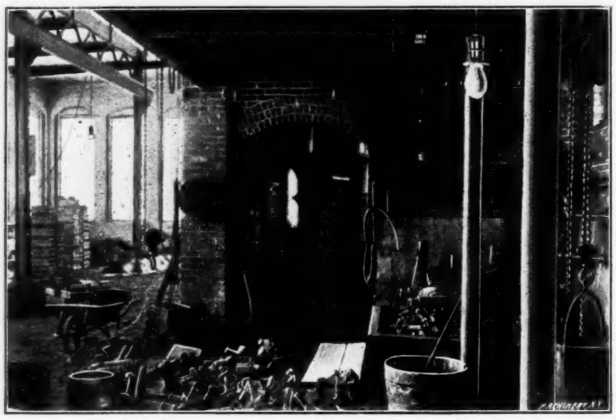
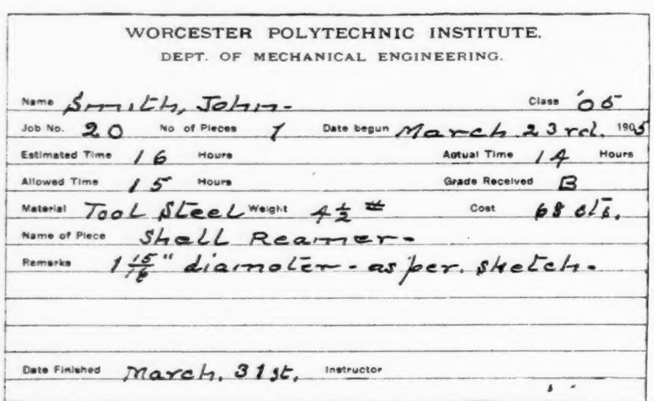


Fig. 9. Another View in the Foundry.



WORCESTER POLYTECHNIC INSTITUTE.
DEPT. OF MECHANICAL ENGINEERING.

Name Smith, John. Class '05
Job No. 20 No of Pieces 1 Date begun March 23rd. 1905
Estimated Time 16 Hours Actual Time 14 Hours
Allowed Time 15 Hours Grade Received B
Material Tool Steel Weight 4½ # Cost 68 cts.
Name of Piece Shell Reamer.
Remarks 1 15/16" diameter - as per sketch.
Date Finished March. 31st. Instructor

Fig. 10. Sample of Time Card given to Student with each Job.

The class is arranged in groups and turns are taken in timing operations. The tabulated report is studied carefully to determine whether or not some of the operations can not be shortened by improving (a) the tools, (b) the machines, (c) changes in the pattern, (d) methods in the foundry. Any changes that promise even a slight improvement are tried or recommended for trial. Jigs, tools, and fixtures designed by the seniors in this work are built by the junior class and are tried out before being placed in the tool room.

Throughout the entire work of the seniors in cost reduction studies, the student is led into taking the initiative and conclusions are arrived at only after a discussion of the subject by the instructor and students. In other words, the student is supposed to feel that the work is his work and that the instructor is a referee or judge to approve or condemn his results only after a proper discussion of matters has taken place.

The work of the second term of the senior year is taken up with the professor of mechanical engineering, and deals with the questions of shop management from the standpoint of the office man. Opportunities are here given the student to learn methods of superintending, and a study of employment questions is made. As the shops have a weekly payroll of good dimensions the student gets practice in making up such payroll and in the different methods that are employed in keeping the time of employees.

A complete cost sheet for some machine is worked out by each member of the class, the data being taken from the actual time cards given in by the journeymen. Fig. 5 shows a lot of sensitive drills as assembled in the shops. Each and every operation in the production of these machines is known to the student and the costs as worked out by him are the actual shop costs which are kept for each lot for the purpose of comparison and improvement.

A study of the duties of the purchasing agent is made and the different price lists and discount sheets discussed. Methods of keeping track of the work as it passes through the shop are taken up, stock lists made out, systems of billing and methods of foreign and domestic shipment are studied, and the student assisted to make as complete an analysis of office methods of superintendence as his time will allow. Here as in all the shop work taken, the line between the engineer and the workman is never lost sight of either by instructor or student, and the whole training is given with the idea that the graduate is to become a master in his line of work, an employer of men, and a director of enterprises.

The responsibility for attendance rests entirely with the students. While they are expected to be present as assigned for their shop practice, they are not required to be there nor to make up lost time. Each student is given an opportunity to work under an instructor in the shops at regular intervals, and he is marked according to the use he makes of these opportunities. If his average for the term falls below 60, he must repeat the work the following year, or else annul his condition by passing a satisfactory examination.

* * *

Tantalum, the metal that has lately been employed by Siemens & Halske for incandescent lamp filaments, or tantalium, as it is variously called, the difference of usage in the spelling of this word being the same as in the case of aluminum and aluminium, is said to possess remarkable possibilities for tool making. N. von Bolton, a chemist, is said to have shown by laboratory experiment that it is both tough and of a hardness almost equal to that of the diamond. A sheet about .039 inch thick was hammered from the first piece produced of the pure metal. Attempts made to drill this by ordinary methods failed; a diamond drill was then used when, after constant work for three days and nights at the rate of 5,000 R. P. M., only one-fourth of the thickness of the sheet had been drilled through, while the drill was so badly worn that the experiment was discontinued. Tantalum is entirely non-magnetic, its fusing point lies about 2,300 degrees Centigrade, and its specific gravity is 14 to 17. In the form of wire it sustains a load of about 128,000 pounds per square inch.

* * *

The *Foundry*, in speaking of strong brass and bronze, says that the Tobin brass alloys are supposed to give over 60,000 pounds in tensile strength. Government specifications for manganese bronze are understood to call for a tensile strength of 72,000 pounds per square inch. Such an alloy is as follows:

	Pounds.
Ingot copper	80
Manganese copper	8
Tin	6
Zinc	6

This will be difficult to cast sound and so will this, which is supposed to be stronger yet:

	Pounds.
Lake copper	96
Silicon copper	4
Aluminum	2

There is no doubt that considerable experimenting will be necessary before a satisfactory alloy for this pressure is found.

THE SPECIFIC HEAT OF SUPERHEATED STEAM

Until quite recently the value almost universally adopted for the specific heat of superheated steam at constant pressure has been 0.48, derived in 1840 from the results of three series of experiments by Regnault, which he considered the most reliable of his investigations upon this subject. The data from this series were:

Specific heat.	Pressure. Lb. per sq. in. abs.	Superheat. Deg. C.
0.48111	49	88
0.48080	33	86
0.47963	31	93

In steam calorimeter work where the temperatures come within the limits of Regnault's experiments, the value of 0.48 is probably correct; but with the high pressures and temperatures now being dealt with in power plants using superheated steam a much higher value should be taken.

In the case of tests upon superheaters and engines or turbines using superheated steam, the weight of the steam does not afford a fair basis for estimating the gain or loss from superheating, because weight alone gives no indication of the amount of heat in superheated steam of a given pressure, as is the case with saturated steam. With our present knowledge, the best way to compare results with and without superheat is by weighing the coal burned under the boiler. But if the specific heat of superheated steam were accurately known, it would be more satisfactory to calculate the efficiency of the apparatus on the basis of the heat units given up to or rejected by the steam at the different steps in the process.

The Importance of a Correct Value for Specific Heat.

The results of some tests will now be given to show to what extent calculated efficiency will vary when using different values for specific heat; also, to compare results when efficiency is calculated in heat units and in pounds of steam per horse-power hour.

The following results are from a test upon a Schmidt superheater reported by Prof. Jacobus in a paper in the proceedings of the A. S. M. E. for 1904.

Superheater Test.

Total dry coal consumed, in pounds	1,426
Heat of combustion of coal in B. T. U., per pound	14,060
Pressure of steam entering superheater, lb. per sq. in. gage	147.4
Temperature of steam entering superheater, deg. F.	365.6
Temperature of steam leaving superheater, deg. F.	809.1
Amount of superheating, deg. F	443.5
Total weight of steam superheated	58,025
Weight of steam superheated per lb. of coal burned	40.69

In the foregoing test the heat of combustion of the coal was carefully determined by calorific tests, and the heat represented by the combustion of each pound of coal is thus known. The weight of steam flowing through the superheater per pound of coal burned, and the amount that it is superheated, are also known. If the specific heat of the steam could be correctly assumed therefore, it would be possible to calculate the amount of heat imparted to the steam per pound of coal burned, from which the efficiency of the superheater could be calculated.

In the table below, the efficiency is calculated on this basis, under the assumption that the specific heats are 0.48 in the first column; 0.6 in the second column, and 0.8 in the third column. For the first value the efficiency is 61.6 per cent; for the second, it is 77 per cent, and for the third it is practically 100 per cent, showing that the specific heat cannot be as high as 0.8. These figures illustrate of how much importance is a correct value of specific heat in calculations upon the efficiency of superheaters.

	Sp. heat = 0.48	Sp. heat = 0.6	Sp. heat = 0.8
Heat imparted to the steam in B. T. U.	12,352,000	15,440,000	20,587,000
Heat imparted to the steam per pound of coal burned in B. T. U.	8,662	10,827	14,436
Efficiency of superheater in per cent, based on heat of combustion of the coal	61.6	77.0	102.7

The reader who wishes to investigate these various experiments should procure a copy of the Journal of the Worcester Polytechnic Institute for November, 1904. It contains an article by Prof. Sidney A. Reeve, reviewing the work of different experimenters and gives references to the original documents where their results are reported.

Turbine Tests.

To show the application of values of specific heat to tests of motors using superheated steam, the following data are given, from tests upon a De Laval steam turbine by Dean and Main. The turbine was tested both with saturated and superheated steam, under similar conditions, so that a direct comparison can be made between its performance when running with saturated and with superheated steam, and the gain from the use of superheated steam determined.

Test with Saturated Steam.

Dry steam entering turbine, pounds per hour	5,052
Initial pressure, lb. sq. in., gage	206.4
Total heat in steam, B. T. U. per pound	1,149.3
Brake horse-power	333
Steam used per brake horse-power per hour, pounds	15.17
B. T. U. per brake horse-power per hour	17,435

Test with Superheated Steam.

Weight steam entering turbine, pounds per hour	4,906
Initial pressure, lb. sq. in., gage	207
Superheat, degrees F.	84
Total heat in steam, B. T. U. per pound—	
When sp. heat is taken = 0.48	1,190
When sp. heat is taken = 0.6	1,200
When sp. heat is taken = 0.8	1,217
Brake horse-power	352
Steam used per brake horse-power per hour, pounds.	13.94
B. T. U. per brake horse-power per hour—	
When sp. heat is taken = 0.48	16,590
When sp. heat is taken = 0.6	16,728
When sp. heat is taken = 0.8	16,965

Under the assumption of specific heats equal to 0.48, 0.6 and 0.8, the total heat contained in the steam and the number of heat units utilized per brake horse-power per hour were calculated as tabulated above. From these the gain from superheating was calculated under the three assumptions, giving the results shown below:

Gain from Superheating.

First, by taking the water consumption in pounds—

$$\frac{15.17 - 13.94}{15.17} = 0.081 = 8.1 \text{ per cent.}$$

Second, by taking the heat units in the steam consumed—

When sp. heat = 0.48

$$\frac{17{,}435 - 16{,}590}{17{,}435} = 0.0484 = 4.8 \text{ per cent.}$$

When sp. heat = 0.6

$$\frac{17{,}435 - 16{,}728}{17{,}435} = 0.04 = 4 \text{ per cent.}$$

When sp. heat = 0.8

$$\frac{17{,}435 - 16{,}967}{17{,}435} = 0.027 = 2.7 \text{ per cent.}$$

The gain from superheating, therefore, calculated on the basis of heat units utilized—which is, of course, the correct basis—is much less than when on the basis of pounds of water per horse-power per hour. In the heat unit calculations the efficiency is seen to vary from 4.8 per cent to 2.7 per cent for the different values of specific heat, showing that the higher the value assumed the less the gain is found to be by using superheated steam.

Results of Tests to Determine Specific Heat.

Many experimenters have attempted to derive values of the specific heat of superheated steam for other temperatures and pressures than covered by the tests of Regnault.

Among the more important work in this connection is that of Grindley in England, of Griessmann and Lorenz in Germany, and of Prof. Carpenter in this country. There are also now in progress at the National Physical Laboratory, Leddington, England, elaborate tests under the direction of the Manchester Steam Users' Association.

The results and conclusions of these various men are in the highest degree contradictory, and it can almost be said that one may arrive at any desired conclusion, or no conclusion at

all, from their work, according to one's method of sifting and sorting the results.

It has been held by authorities in thermodynamics that the specific heat of superheated steam at constant pressure is a function of the temperature but not of the pressure. On the basis of certain assumptions, which, however, are probably not correct, it can be demonstrated mathematically that such should be the case. From actual tests, however, the specific heat is found to be a function both of the temperature and of the pressure. A review of Grindley's results has been made by Prof. Reeve, of Worcester Polytechnic Institute, who, by recalculation of the values, concludes that they depend both upon temperature and pressure. He publishes tables giving the values of specific heat and also of the total heat of superheated steam for a limited range of temperatures and pressures, in his article in the *Journal of the Worcester Polytechnic Institute* previously referred to in the footnote.

One of the most useful reviews of the work of the different experimenters was contributed by Mr. George A. Orrok, chief draftsman of the New York Edison Co., to *Power* for August, 1904. He plotted the various values and found those of Griessmann to be the most consistent, and also to agree quite closely with those of Grindley, as recalculated by Reeve. Taking the Griessmann values as a basis Mr. Orrok deduces the following formula to represent them:

$$C_p = 0.00222\ t_s - 0.116$$

in which t_s = temperature of the superheated steam.

The above formula gives the instantaneous or true specific heat at any temperature. The *mean* value of the specific heat between the points of saturation and any degree of superheat can be found by the formula

$$C_p = 0.00222\left(\frac{t_s + t}{2}\right) - 0.116$$

$$= 0.00111\ (t_s + t) - 0.116$$

To determine the total heat of superheated steam we have

Total heat $= \lambda + C_p\ (t_s - t)$ where

λ = total heat of saturated steam at the given pressure.

t_s = temperature of the superheated steam.

t = temperature of the saturated steam at the given pressure.

C_p = specific heat of the steam at constant pressure.

The value of λ is given in steam tables.

To illustrate the application of Orrok's formulas we will take the following data:

Steam pressure.................. 100 pounds absolute
Temperature of superheated steam....... 450 deg. F.

From the steam tables we obtain

$$t = 327.58$$
$$\lambda = 1{,}181.9$$

To calculate the instantaneous or *true* specific heat

$$C_p = 0.00222\ t_s - 0.116$$
$$= 0.00222 \times 450 - 0.116$$
$$= 0.999 - 0.116$$
$$= 0.87$$

To find the *mean* specific heat between saturation and the temperature 450 degrees of the superheat, we have

$$C_p = 0.00111\ (t_s + t) - 0.116$$
$$= 0.00111 \times 777.58 - 0.116$$
$$= 0.897 - 0.116$$
$$= 0.75$$

To find the total heat of superheated steam at 450 degrees temperature and 100 pounds pressure, we have

$$\text{Total heat} = \lambda + C_p\ (t_s - t)$$

in which C_p is the mean specific heat between the points of saturation and of the temperature of 450 degrees = 0.75, as found above. Hence

$$\text{Total heat} = 1{,}181.9 + 0.75\ (450 - 327.58)$$
$$= 1{,}181.9 + 91.8$$
$$= 1{,}273.7$$

To show the closeness of Grindley's results, as recalculated by Reeve, and of Griessmann's values, as given by Orrok, we have compared below several values for total heat, taken at random from Reeve's table, with corresponding values calculated by using Orrok's formula for specific heat. The agreement is seen to be close, and as Orrok's formula for specific heat is a simple one to use, it apparently is as satisfactory as any that we have.

COMPARISON OF TOTAL HEATS, CALCULATED IN ACCORDANCE WITH RESULTS FROM GRINDLEY'S AND GRIESSMAN'S TESTS, AS PRESENTED BY PROF. REEVE AND GEORGE A. ORROK.

Absolute Pressure.	Temperatures of Superheated Steam.	Total Heat of Superheated Steam according to Reeve.	Total Heat of Superheated Steam according to Orrok.	Difference between the two Values.
30	275	1,169.3	1,169.8	0.5
30	320	1,194.88	1,194.3	0.58
51.5	300	1,176.86	1,177.4	0.54
51.5	330	1,195.4	1,194.9	0.5
78.5	330	1,187.61	1,188.2	0.59
124.5	355	1,194	1,194.2	0.2
149.5	364	1,195.07	1,195.2	0.13

In 1901 Prof. Lorenz undertook an extensive series of experiments upon the specific heat of steam, extending over a wide range of pressures and temperatures. His work was under the direction of the *Vereines Deutscher Ingenieure* and is reviewed by Robert H. Smith in the *Engineer* (London), July 8, 1904. The peculiarity of the results of Prof. Lorenz is that while they indicate an increase of specific heat with the pressure, they also show a decrease with increase of temperature. Further than to make the above general statement Prof. Lorenz does not attempt to deduce any law. He remarks that for low pressures Regnault's value of 0.48 seems to hold good, while for high pressures 0.6 is approximately correct. An inspection of the table published in the *Engineer* shows that a large increase in temperature, accompanied by a slight increase in pressure, can take place without any change in the specific heat, owing to the opposite effects of the temperature and pressure upon the specific heat, as determined by Prof. Lorenz.

It now remains to refer to the tests conducted under the direction of Prof. R. C. Carpenter, at Cornell University. Experiments began in 1891 and were carried on at different times over a period of several years. The later tests were made by Prof. C. R. Jones of West Virginia University, who was taking a post-graduate course at Cornell. As a result of the latter's work Prof. Carpenter has published the following values:

Absolute Pressure.	Specific Heat.	Absolute Pressure.	Specific Heat.
14.7	.484	80	0.563
20	.492	100	0.614
40	.523	120	0.645
60	.553		

These results may be expressed by the equation

$$C_p = 0.462 + 0.001525\ p.$$

where p is the absolute pressure.

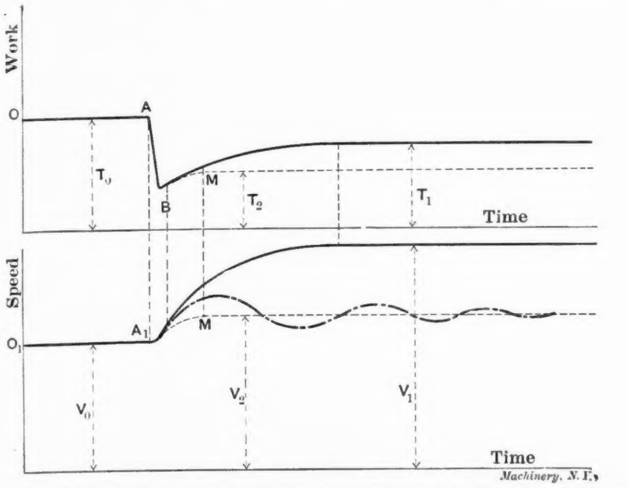
The apparent specific heat, therefore, at constant pressure, as deduced by Profs. Carpenter and Jones, is 0.484 at atmospheric pressure, and increases with the increase of pressure, becoming about 0.645 at 120 pounds absolute. Prof. Carpenter states that if there is any increase of specific heat with increase of temperature for a constant pressure, it is so slight as to be obscured by errors of instruments or of observation in the experiments, in which the degree of superheat ranges from 90 degrees to 30 degrees in excess of the temperature of saturated steam.

In view of the contention of some writers in thermodynamics that specific heat at constant pressure of superheated steam is a function of the temperature only, the Sibley College results are of unusual interest, and the writer has corresponded with Prof. Jones in regard to the matter, who replies that with the apparatus in operation he thinks it would not require very much time to convince any engineer that the specific heat does increase with the pressure. He states that his experiments with varying temperatures were made with steam at 20 pounds gage pressure and at temperatures running from the normal temperature of saturated steam to about 400 deg. F. The higher and lower temperatures in each experiment were kept about 10 degrees apart, that is, steam would be taken in at 300 degrees, say, and discharged at 290 degrees. When the tests were finished it was supposed that there were sufficient data to draw some definite conclusions in regard to the variation of the specific heat with the temperature; but when the final results were obtained the variations were found to be so slight that no conclusion in regard to the subject could be deduced, after making corrections for radiation.

THE BOUVIER GOVERNOR FOR WATER TURBINES.

DR. ALFRED GRADENWITZ.

For the consideration of the effect of a regulator for water turbines the diagrams given in figures 1 and 2 are drawn, representing work and speed respectively to a base line of time. If now a turbine rotating at a uniform speed V_0 and driving any kind of machine which absorbs an amount of energy T_0 at the given speed, be considered, a straight line O A, Figure 1, representing the resisting energy T_0, and the line $O_1 A_1$ (Figure 2), representing the uniform speed V_0, will be obtained. If now at a given moment corresponding to the points A and A_1, Figs 1 and 2, the resisting energy T_0 be altered suddenly



Figs. 1 and 2. Diagrams showing Variations of Work and Speed of a Water Wheel under different conditions for a given time.

without the aperture of the turbine being varied, a change of speed will be produced, and after a certain time the speed will have become V_1, such variation of speed obviously being greater as the variation in load has been greater. This variation of speed takes place with an acceleration which gradually diminishes, becoming zero when the whole set possesses its new normal speed. Letting A B in Figure 1 represent a sudden diminution of the resisting work, the curve of speeds and the curve of resisting work will assume the shapes shown in full lines in figures 1 and 2; the new values T_1 and V_1 being the resisting work and speed corresponding to the new conditions. If, however, a regulator acting on the gate mechanism of the turbine be used, the equilibrium between the motive and re-

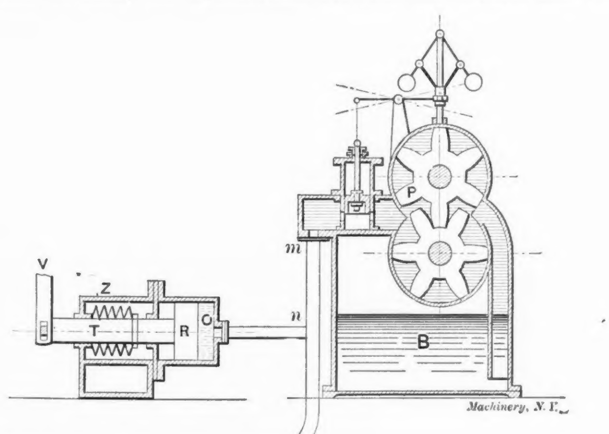


Fig. 4. Diagram of the Bouvier Governor. The Pipe at the bottom may be in communication with other Regulating Cylinders.

sisting forces will be reached much more rapidly, supposing the same variation to take place in the resisting work, and the speed curve will assume the shape shown in dotted lines in Fig. 2 up to the point M, where equilibrium between these forces is obtained. If at this moment the action of the regulator could be discontinued, another normal speed V_2 would be obtained, this speed being less different from V_0 than was V_1. For a given variation of the resisting work it is always possible thus to so calculate the moving masses and the speed of working of the regulator that the difference V_2-V_0 shall not be beyond a given limit. The ideal condition for the working of a regulator would thus consist in a very rapid action on the locking gates of the turbine, this action ceasing completely as the motive forces compensate the resisting forces.

In practice up to this time there has always been a certain "hunting" of the governor. The governor has not been designed to act immediately on the water supply, but has been confined to stopping or starting an auxiliary motor, the energy of which is derived either from the turbine itself or from the water submitted to pressure. The connection between the governor and the auxiliary motor was formerly a simple engaging or disengaging device. With this arrangement the movement of the slide gate will not cease exactly at the moment of equilibrium between the forces, the oscillations thus resulting from the repeated action of the governor being similar to those represented in the dot and dash curve in Fig. 2.

These oscillations are avoided in an apparatus recently patented by A. and H. Bouvier, Grenoble, France, in which any disturbance occurring in the resisting work is immediately corrected by a corresponding displacement of the gate mechanism, so that another state of equilibrium follows immediately upon the intial state. This apparatus is illustrated in Fig. 4. The liquid is pumped through an orifice by a pump P of a practically constant output, the amount of opening of this orifice being altered by the displacement of a piston, whose position is controlled by a fly ball governor. For every position of the piston there will eventually be a given pressure in the discharge pipe of the pump, this pressure being inversely proportional to the area of the valve opening at any moment for a constant output. If the speed of the governor is increased the aperture of the opening will diminish progressively, as the

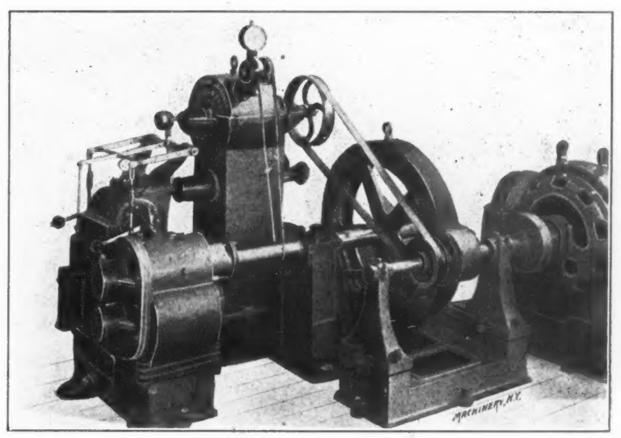


Fig. 3. Details of the Central Regulating Apparatus in the Bournillon Power House.

pressure augments. On the speed becoming constant the governor socket cases to rise, so that the discharge pressure of the pump ceasing to increase, retains a given value that corresponds to the position of the slide. The pressure produced in the discharge pipe acts on a piston R in Fig. 4, this piston being counteracted by a spring Z; the piston R occupies a perfectly determined position for each value of the pressure of the water pumped. The piston R being rigidly connected to the regulating slides of the turbine there can be for a given position of the governor at any moment but a single possible position of the locking gate.

In operation, if the motor rotating at a uniform angular speed V_0 is suddenly relieved of part of its load the angular speed will augument, the governor rise and the valve diminish the aperture of the discharge opening, while the pressure augmenting, the spring will be compressed and the sluice gates be closed more and more. This action of the gate mechanism, however, ceases at the very moment when the motive forces have become equal to the resisting forces, when the pressure becoming constant the compression of the spring ceases, and the whole system is in another state of equilibrium at an angular speed V_2, a little superior to the initial speed V_0. If, however, the resisting work had increased suddenly, the speed would have diminished and the pressure of the liquid become less, so that the spring being expanded any displacement would have been stopped as soon as the diminution of speed had ceased. The variation of speed V_2-V_0 can be made as small as desired for a given variation of load with this governor, if

the moving masses are properly calculated. A simple contrivance allows the speed to be restored from the speed V_2 to the initial speed V_0 if it should be desired to keep the speed at accurately constant figures.

Where too great an amount of energy would be consumed in the displacement of the gate mechanism if the piston *R* acted directly on this mechanism as described, the apparatus is completed by an additional hydraulic motor constituting a relay, so that the piston *R* has only to control the distribution valve of this motor, the piston of the motor following exactly

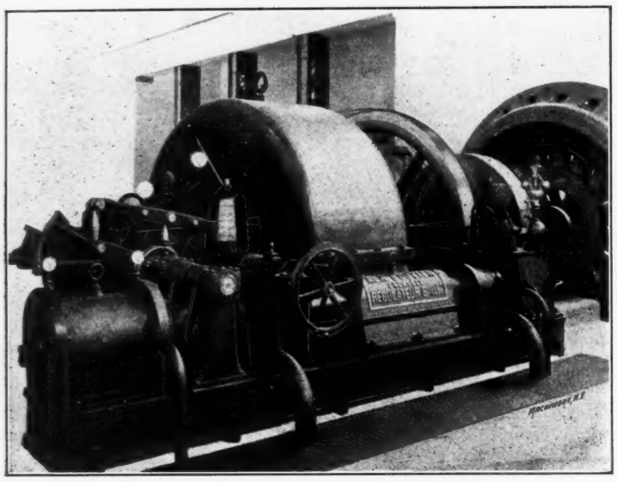


Fig. 5. A single Electro-hydraulic Generating Set with its Regulator at the Bournillon Power House.

and instantaneously any displacement of the valve. The motor can be worked with the motive water if there is a sufficient head available, there being no sensitive parts. If the fall is not sufficient the motor is actuated by a liquid under pressure, derived from an accumulator fed by pumps. With this additional motor it is of course the piston of the motor which controls the locking gates of the turbine.

The apparatus is adapted for the simultaneous regulation of a set of hydraulic motors driving alternators connected in parallel, each motor being fitted with a pressure cylinder and springs *Z*, like that shown in Fig. 4. The pressure produced

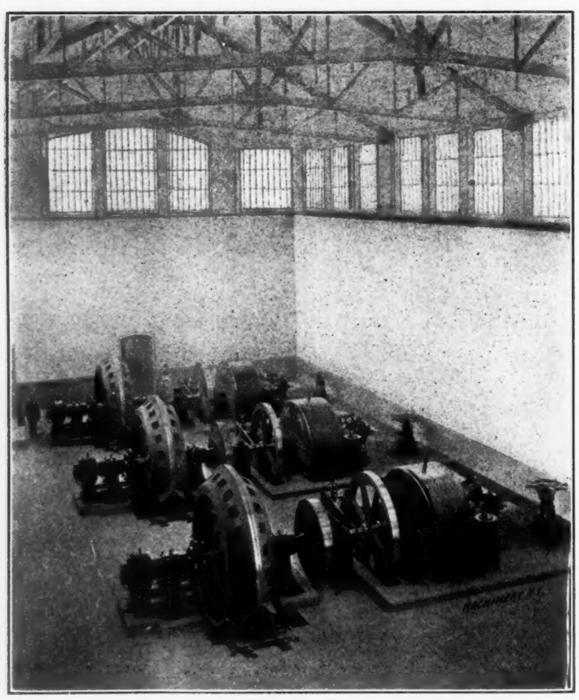


Fig. 6. Three Generator Sets with one Central Regulator, Bournillon Power House.

by the pump *P* in Fig. 4 may be transmitted simultaneously to the locking devices of each of the motors, and if these motors be exactly similar and the springs *Z* of the various regulating devices be adjusted similarly, the locking gates of the turbines will all be in the same position for the same pressure of water in the various pressure cylinders and the alternators will hence be put to equal loads. If the turbines have different outputs the various springs *Z* will have to be so adjusted that the motors always work at the same fractions of their output. In central stations a single centrifugal force governor will be sufficient to regulate the speed of all the motors, as shown in Fig. 6, which represents three generator sets with one central regulator as applied by Messrs. Bouvier in a recent hydro-electric installation at Bourillon (Isère). This plant being intended for supplying the town of Vienne (Isère) with power and light, includes three turbines with 1,250 H. P., making 375 revolutions per minute, under a head of 98 meters. The machine shops are, however, to be fitted with three additional sets of 1,250 H. P. working under a pressure of 300 meters.

The central regulator installed according to the above system controls the locking devices of the three turbines at present installed, as shown in Fig. 6. The details of the central apparatus are shown in Fig. 3, and a view of a single set with its regulator in the Bournillon power house is disclosed in Fig. 5. The official tests made on the delivery of the turbine proved the small variation of speed allowed by the regulator. From one of these tests we see that by cutting out instantaneously

a load of 500 H. P. a variation of speed of 6 revolutions;
a load of 670 H. P. a variation of speed of 9 revolutions;
a load of 900 H. P. a variation of speed of 12 revolutions;
a load of 1,000 H. P. a variation of speed of 13 revolutions,

has been recorded, the normal angular speed of the turbine being 375 revolutions per minute and its maximum output 1,250 H. P.

* * *

BALANCING MOVING PARTS.

One bad feature of the slotter as ordinarily constructed is the counterweight. The vertical travel of the ram makes it necessary to balance this weight. In most machines this is effected by a pivoted arm connected with the ram at one end and carrying a counterweight at the other. In order to save weight in the counterweight, it is the almost universal practice to make the counterweight arm longer than the distance from the pivot to the ram. Now while it is possible to effect a standing balance with this construction, it is not possible to secure a dynamic balance. In other words a standing is not the same as a running balance. The faster the slotter works, the greater is the difference in unbalanced forces, since the kinetic energy of the mass of the ram and counterweight both increase with the square of the velocity. With its long arm the counterweight must necessarily increase in kinetic energy faster than the ram, hence when working at high speeds there is a noticeable lack of balance in slotter action. Properly constructed the counterweight arm should be of the same length as the arm connecting the ram, and then no matter at what speed the ram is operated, the balance will be unchanged. A type of slotter meeting this requirement is that having the ram balanced by a wire rope or chain running over pulleys and connected to a counterweight on the opposite side. And, in general, the balancing of a machine should be accomplished, if possible, by the use of counterweights of the same weight as the parts to be balanced, and moving at the same velocity.

* * *

Frank W. Mahin, Consul at Nottingham, England, writes that the Elektricitäts-Aktien-Gesellschaft, of Frankfort, has recently introduced a machine for testing the lubricating qualities of oils. The essential part is a short shaft working in a bearing, and loaded appropriately. About half a pint of the oil under examination is poured on the bearing, and the shaft is set revolving at a definite speed. The time that elapses before the shaft comes to rest is noted; the greater the time the better is the lubricating quality of the oil. After the test the bearing is cleaned by pouring over it a liquid in which the oil is soluble, and then removing the liquid by a blast of air. This method of cleaning is found to be quite effective and is economical of time. The machine may be driven by an electric motor or other mechanical means or by hand, and there is an arrangement of resistance coils by which the bearing can be heated up to any required temperature. Both the bearing pressure and the speed may be conveniently regulated.

AN EXAMPLE OF WORM GEARING.

OSCAR E. PERRIGO.

It is a well-known fact that the power required for boring large holes in hard metals is a severe test to any arrangement of gearing designed for the purpose, particularly when rapid work is expected to be accomplished, and that it is a matter of congratulation to the builder when such mechanism successfully withstands the strains incident to the hard usage to which it is necessarily subjected. This is still further enhanced when the mechanism successfully meets these rigid requirements year after year with no apparent deterioration, or expense for the repairs usually expected in this class of machines.

A case in point which well illustrates these features is the heavy boring lathe, a rear elevation of which is shown in Fig. 1. This lathe has a headstock heavily back-geared and provided with a three-step driving cone, for an extra wide belt, thus giving it an abundance of driving power for all work expected of it. The bed is extra heavy and deep, so as to resist the heavy strains put upon it by the strongly geared feeding mechanism, and was built considerably longer than is shown in the engraving, and provided with a center support.

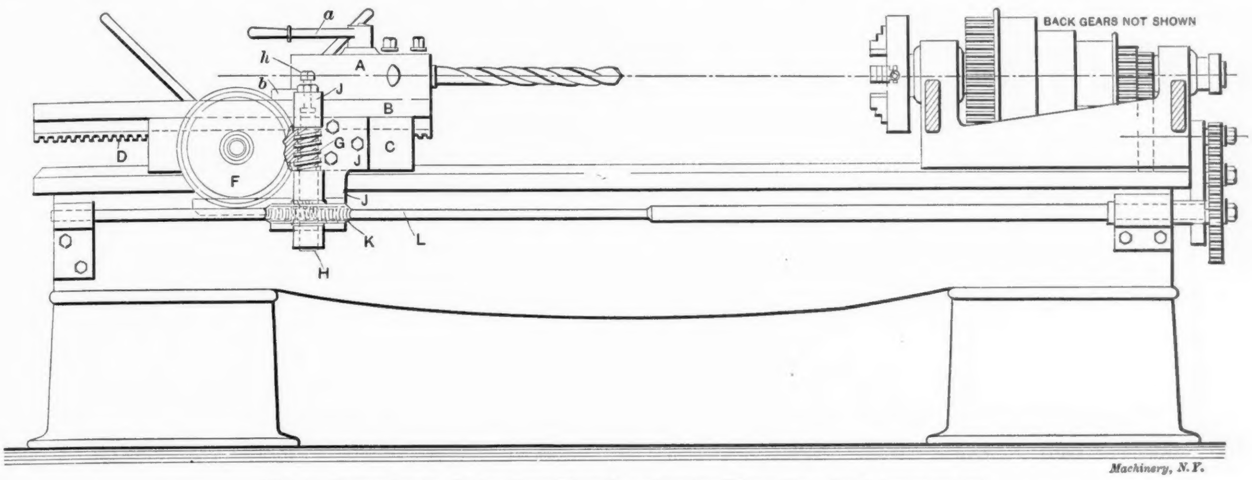


Fig. 1. Heavy Boring Lathe, Rear Elevation, showing Worm Gear Feed Arrangement.

The boring tools are carried by a 15-inch turret, *A*, pivoted upon a heavy top slide, *B*, four feet long, so as to give a travel of three feet. This slide is in turn supported upon a substantial base, *C*, secured to the bed at any desired point by clamps operated by four eccentrics (not shown). The turret is clamped in place by the hand lever, *a*, as usual, and has, for additional security, a projecting flange, *b*, provided with grooves opposite each of the tool holes, in which engages a securely pivoted lever (not shown).

The feed arrangement is constructed as simply and with as few pieces as possible. Upon the under side of the slide, *B*, is secured a wide steel rack, *D*, of coarse pitch, and engaging a cast steel pinion on the shaft upon the outer end of which is fixed the main worm wheel, *F*, which is engaged by the main worm, *G*. This worm is fixed upon a vertical shaft, *H*, journaled in the bracket box, *J*, bolted to the base, *C*. Upon the lower end of the vertical shaft, *H*, is fixed the secondary worm gear, *K*, engaging the secondary worm carried by the feed shaft, *L*, which is driven by a series of change gears at the headstock in the usual manner. The worm gears, *F* and *K*, are of the best quality of close-grained cast iron and the worms and their shafts of 40-point carbon steel. To provide for the severe upward thrust of the worm shaft, *H*, an anti-friction washer is placed over it and held down by a hardened steel screw, *h*, secured by a check nut, as shown. Beneath the worm on the feed shaft, *L*, and also under the main worm wheel, *F*, are pockets for holding a quantity of heavy oil suitable for the use of worm gearing. With the exception of being of unusual strength the parts are of ordinary construction and arrangement except the worm gears.

Many good mechanics are prone to object to any kind of a worm gear, and can cite numerous examples wherein they have proven failures and utterly worthless for the purposes intended, until there is a very strong prejudice against them in any form. The writer is of the opinion that there is really only one practical objection to a properly constructed worm gear, and that is, it must be constantly lubricated, and men running machines in which they are used are very liable to forget this fact altogether.

The principal, and almost the only reason why worm gears fail to give satisfactory results is that usually they are not properly designed at first. Another is that they are not properly hobbed out, and sometimes not hobbed at all. It is the purpose of this article to point out how they should be designed in order that they may be successful, and to present the machine shown in Fig. 1, as an example of their practical utility when properly constructed.

There are various methods for determining the diameter of the pitch circle of a worm gear. One authority takes the outside diameter of the turned blank at its smallest diameter, or throat, as proper. Another takes the diameter of the bottoms of the teeth at the extreme edge of the cut gear. Still another, the point where the pitch line of the worm intersects the center line passing through the worm and worm gears. All of these are more or less in error, as they do not take proper account of the width of the face of the gear. If the teeth are straight, as in a spur gear, we naturally take a point in the center of the teeth (after subtracting the clearance) as the pitch line. Now when we have a curved tooth it is obviously not proper to do this, as the actual working pitch diameter must be somewhat larger than this. But how much larger should evidently be determined by the amount of contact with the worm, that is, the angle within which this contact is to be, the width of face being in turn controlled by the diameter of the worm.

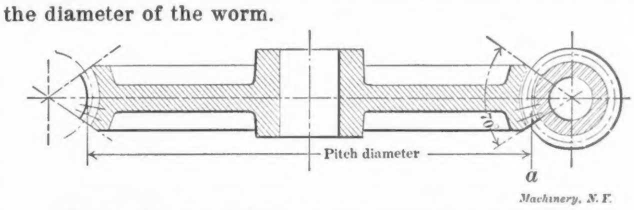


Fig. 2. Method of Determining Pitch Diameter of Worm Gear.

Practically the face of the worm gear is about equal to one-half the outside diameter of the worm, but the matter is best considered by saying that the inclosing angle of contact should not be less than 45 degrees nor more than 80 degrees, while from 60 degrees to 70 degrees will be found most useful. The writer has found by ample practice that the true working pitch diameter is most nearly determined by the method shown in Fig. 2, which represents a worm wheel having a contact of 70 degrees. To determine the pitch diameter, divide the arc of the pitch line of the worm, contained between the center line and one of the lines of the enclosing angle, into three equal parts, and draw the line, *a*, at the intersection of the second line from the center line. This will give the point from which to measure the pitch diameter. If this is laid out on a large scale and with various angles of contact, the difference between it and the usual methods will be more clearly shown than it is in the engraving, and it will be found to make from

two to four teeth difference from those methods in a worm wheel of, say 12 inches diameter.

All worm gears should, of course, be hobbed out, and the thread of the worm made to carefully conform to that of the hob. The sides of the thread are usually of a 15-degree angle.

Now as to the proof of the correctness of this method, which the writer has used successfully for years. The machine shown will readily bore 3-inch holes in 50-point carbon steel spindles. In several cases where a 5⅛-inch hole was required, it was first bored 2 inches and then a boring bar, provided with two double-end cutters, was introduced, enlarging the hole from 2 inches to 5⅛ inches at one cut and taking out nearly thirty pounds of chips per hour.

One end of the spindle to be bored is held in the chuck and the other end supported by an ordinary center rest. Oil or drilling compound is forced through a tube to the point of the drill, or the cutters, lubricating the cutting edges and washing out the chips.

This machine has been in use for over seven years, and the same worms and worm gears are on it that were put on when the machine was first built, and they are in good condition for as many years more of good service. The working faces do not seem to have changed their original form during the entire time, which is taken as ample evidence that they were right originally, as the writer has frequently seen worm gears in lathe aprons, designed after the usual methods, entirely worn out with six or eight months' service.

* * *

NOTES ON THE STRENGTH OF BEAMS, PLATES AND COLUMNS.

R. P. KING.

Some time ago I noticed in your columns an explanation of a simple method of finding the strength of beams, using moments. I had in mind at that time writing a letter on this subject, but have not until lately had time to do so.

It has always seemed to me that beam calculations were simpler by the method of the moment of resistance than by finding the bending moment. Fig. 1 is the table to be used in connection with this method. It differs from the method by moments in that the sign M, the bending moment, does not figure at all, its place being taken by the moment of resistance, and the weight or load, W, comes out of the formula directly. The values of f are the ones usually found and are on the sheet merely to make the whole complete for office use. A table explaining the notation will be found further along.

As to the derivation of this table, take for example Case 7. As usually given, $M = \frac{W\,l}{2}$, in which l is expressed in inches. Substitute for the bending moment, M, the moment of resistance $S\,Z$ and let L = length in feet. Then

$$S\,Z = \frac{12\,W\,L}{2} \text{ or } W = \frac{2\,S\,Z}{12\,L}$$

In looking through these formulas one notices a similarity, or even more than a similarity. In the table below, the points of difference are set forth. Let $a = \frac{S\,Z}{12\,L}$, then in

Case 1 $W = a$.
Case 2 $W = 4\,a$.
Case 4 $W = \frac{16}{3}\,a$.
Case 5 $W = 8\,a$.
Case 7 $W = 2\,a$.
Case 8 $W = 8\,a$.
Case 9 $W = 8\,a$.
Case 10 $W = 12\,a$.

In the handbooks on structural steel are tables for different sections used as beams, supported at both ends, and uniformly loaded, conforming to Case 8. By referring to the above table it will be readily seen that in order to substitute Case 1 for Case 8, we have only to divide the safe load given in the handbook for the section and span in question, by 8.

The factor of safety given in these tables may also be changed to conform to other conditions, the factor in the table being 4. For instance, suppose a case where we have a cantilever beam conforming to Case 7, but the load is a live load and we wish to use a safety factor of 8. We see that the safe load varies as 2 is to 8, and the safety factor varies as 4 is to 8, so all we have to do is to multiply the load calculated for case 7 by 8, i. e., $\frac{8}{2} \times \frac{8}{4}$, and in the book we find the most economical section for this load for the given span.

When designing machinery, it is quite often necessary to know how to determine the section modulus quickly. In order to do this we must first find the neutral axis and the moment of inertia. The neutral axis is that portion of the beam where no bending takes place and is usually assumed to be at the center of gravity of the section. While this is probably not rigorously true in any case, it may be assumed as sufficiently so for all practical purposes.

If a section has an axis of symmetry, the center of gravity is somewhere on this line. If it has two axes of symmetry the C. G. is at the intersection. If the section has one axis, draw a line perpendicular to that axis. Divide the section into small figures the C. G. of which may be easily found, rectangles being the ideal figures. Multiply each area by the distance from its C. G. to the perpendicular line and divide the sum of these products by the entire area of the section.

If the section has no axis, draw two lines at 90 degrees and determine the C. G. with reference to one of the lines as directed above; then with reference to the other, and the C. G. will be at the intersection of the two resultant lines.

Table showing Notation used in the Formulas.

L = Length in feet.
l = Length in inches.
I = Moment of inertia.
E = Modulus of elasticity.
W = Load in pounds.
S = Stress in pounds per square inch.
Z = Section modulus.
M = Bending moment.
f = Deflection in inches.
r = radius of gyration with reference to a neutral axis perpendicular to the plane of flexure.
d and D = Diameters in inches.
p = Pressure pounds per square inch.
t = Thickness in inches.
K = Constant.
C and C' = length in inches.

To determine the moment of inertia, divide the section into many small parts and multiply the area of each part by the square of the distance from its C. G. to the neutral axis; the sum of these products will be the moment of inertia. The results by this rule will always be a little small.

We are now ready for the section modulus. Divide the moment of inertia of the cross section of a beam by the distance from the neutral axis to the extreme fiber, i.e., the fiber that is farthest from the axis, and the quotient will be the modulus of the section.

The modulus of ordinary sections is given in all pocket books. The section modulus for rectangular sections is $\frac{b\,d^2}{6}$, where d = depth of beam and b = breadth. This quantity is used in calculating timber beams, among other things, and I want to call attention to the ease with which these beams may be varied. For instance, a beam 6 x 8 placed edgeways, was found to be too light for a certain load. As the beam varies in strength directly as its breadth and as the square of its depth it is easily seen that a beam 8 x 8 would support one-third more, or the loads could be in proportion as 6 is to 8. A beam 6 x 16 placed edgeways would hold four times the load held by the 6 x 8. The strength of round beams varies as the cube of the diameter and nearly all regular shapes will vary in some such way.

The strength of a beam also varies in inverse proportion to its length, so that a beam that is safe under a given load will support twice that load if the length of the beam is one-half. There are two exceptions to the above, one when the load is so great that the beam is crushed, and one where the

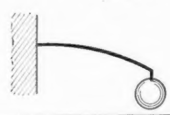
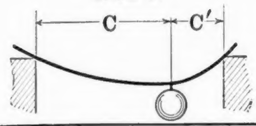
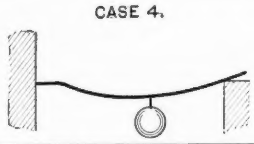
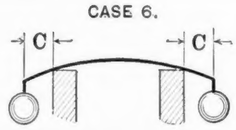
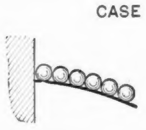
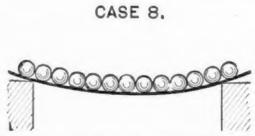
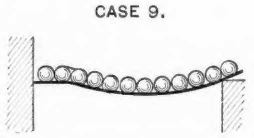
beam fails from shearing off at the supports. Under ordinary circumstances, neither of these will occur.

The treatment of composite beams is more complex, yet may be reduced to a simple form that is easily calculated and is approximately correct. These beams are of steel and cast iron, steel and wood, or steel and concrete, and should be so designed that the steel alone is to take the tension, while the other material is depended upon to take the compression.

may further simplify by assuming that the breadth of the beam is one inch. To figure the depth of the beam, suppose that we forget for the time being that any steel is to be used, and calculate the beam by the formula of Case 8, Fig. 1,

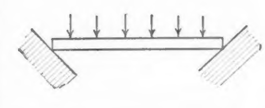
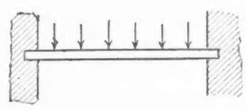
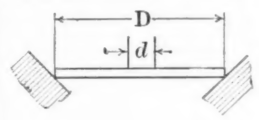
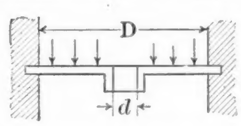
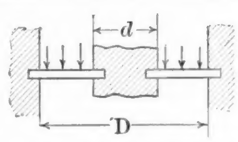
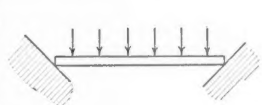
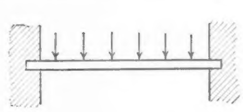
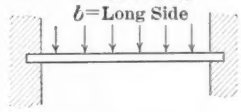
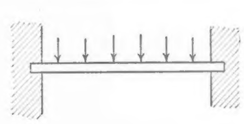
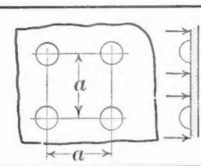
$$W = \frac{8\,S\,Z}{12\,L} \text{ where } Z = \frac{b\,d^2}{6}.$$

If the compressive strength of the concrete is 600 pounds per

CASE 1. 	CANTILEVER. LOAD CONCENTRATED.	$W=\frac{SZ}{12L}$ $f=\frac{(12L)^3 W}{3IE}$
CASE 2. 	SUPPORTED BOTH ENDS. LOAD CONCENTRATED.	$W=\frac{4SZ}{12L}$ $f=\frac{(12L)^3 W}{48IE}$
CASE 3. 	SUPPORTED BOTH ENDS. LOAD CONCENTRATED AT ANY POINT.	$W=\frac{12LSZ}{CC'}$ $f=\frac{(12L)^3 W}{3IE} \times \frac{C^2}{L^2} \times \frac{(C')^2}{L^2}$
CASE 4. 	FIXED AT ONE END, SUPPORTED AT OTHER. LOAD CONCENTRATED.	$W=\frac{16SZ}{3(12L)}$ $f=\frac{W}{IE} \times \frac{(12L)^3\,7}{768}$
CASE 5. 	FIXED AT BOTH ENDS. LOAD CONCENTRATED.	$W=\frac{8SZ}{12L}$ $f=\frac{W}{IE} \times \frac{(12L)^3}{192}$
CASE 6. 	LOADS OVERHUNG. BOTH LOADS CONCENTRATED	$W=\frac{SZ}{C}$ at each End $f=\frac{W}{IE} \times \frac{(12L)^3}{8} \times \frac{C}{(12L)}$
CASE 7. 	CANTILEVER. LOAD UNIFORM.	$W=\frac{2SZ}{12L}$ $f=\frac{W}{IE} \times \frac{(12L)^3}{8}$
CASE 8. 	SUPPORTED BOTH ENDS. LOAD UNIFORM.	$W=\frac{8SZ}{12L}$ $f=\frac{W}{IE} \times \frac{5(12L)^3}{384}$
CASE 9. 	FIXED AT ONE END, SUPPORTED AT OTHER, LOAD UNIFORM.	$W=\frac{8SZ}{12L}$ $f=\frac{W}{IE} \times \frac{(12L)^3}{192}$
CASE 10. 	FIXED AT BOTH ENDS. LOAD UNIFORM.	$W=\frac{12SZ}{12L}$ $f=\frac{W}{IE} \times \frac{(12L)^3}{384}$

Machinery, N.Y.

Fig. 1. Table of Beam Calculations.

	ROUND PLATE	$t=d\sqrt{\frac{5}{24}\frac{p}{s}}$
	ROUND PLATE	$t=d\sqrt{\frac{p}{6s}}$
	ROUND PLATE $t=K\sqrt{\frac{p}{s}}$	When $\frac{D}{d}=$ 10, 20, 30, 40, 50; $K=$ 1.138, 1.266, 1.327, 1.373, 1.407
	ROUND PLATE $\frac{d}{D}=$ 0.1, 0.2, 0.3; $K=$ 0.402, 0.386, 0.324	$t=K\,d\sqrt{\frac{p}{s}}$
	ROUND PLATE $\frac{d}{D}=$ 0.3, 0.4, 0.5; $K=$ 0.276, 0.225, 0.182	$t=K\,d\sqrt{\frac{p}{s}}$
	SQUARE PLATE a=Side of Square	$t=0.4796\,a\sqrt{\frac{p}{s}}$
	SQUARE PLATE a=Side of Square	$t=\frac{a}{2}\sqrt{\frac{p}{s}}$
a=Short Side b=Long Side 	RECTANGULAR PLATE a and b=Sides of Rectangle	$t=0.707\,a^2 b\sqrt{\frac{p}{(a^4+b^4)s}}$
	ELLIPTIC PLATE a = Major Axis b = Minor Axis	$t=\frac{b}{2}\sqrt{\left(\frac{2a-b}{a}\right)\frac{p}{s}}$
	STAYED PLATE a=℄ to ℄ Staybolts	$t=0.417\,a\sqrt{\frac{p}{s}}$

Machinery, N.Y.

Fig. 2. Formulas for the Strength of Flat Plates.

In every beam the total compression equals the total tension, otherwise the beam would fail. Let us start with this condition and endeavor to find a simple way to calculate a reinforced concrete steel beam for a given load at a given span. The form of such a beam will be a rectangle and we

square inch and we assume the tensile strength to be the same (which it is not), we can calculate the depth of the beam to be O inches, Fig. 3.

We will now proceed to put in the steel. In most cases it will be found about right to make $n = \frac{1}{6}\,O$. In any case the

tension = area of the steel bars per unit $b \times m \times$ the strength of the steel. This may be put into a formula, but is quite as easily handled as given above.

For the strength of columns, we use Gordon's formulas, adapted from Hodgkinson's, and which are given in Fig. 4. In these formulas the values of p are per square inch of section

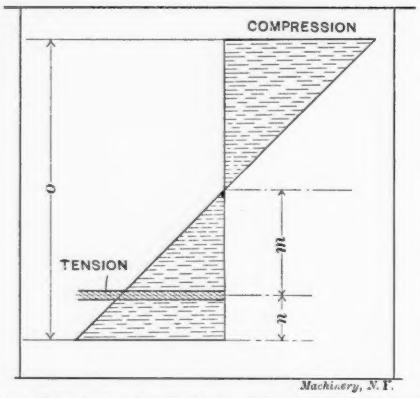


Fig. 3. Calculation for Reinforced Concrete Steel Beam.

and are the ultimate stresses. To find the safe load for a given section it is necessary to multiply p by the area of the section and divide by the factor of safety.

These formulas seem more epecially adapted to building and bridge design, but if we consider a piston rod as a column with square bearings, and a connecting rod a column with pin bearings, their adaptation to machine design becomes apparent. Perhaps the term compression bars would suit this subject better than columns. In these formulas we have r, the radius of gyration. To determine the square of the radius of gyration, it is only necessary to divide the moment of inertia by the area of the section. The value of r^2 is, however, given in the hand books for all ordinary sections.

STEEL COLUMN SQUARE BEARINGS	$p = \dfrac{50{,}000}{1 + \dfrac{(12\,L)^2}{36{,}000\,r^2}}$
STEEL COLUMN ONE SQUARE AND ONE PIN BEARING	$p = \dfrac{50{,}000}{1 + \dfrac{(12\,L)^2}{24{,}000\,r^2}}$
STEEL COLUMN PIN BEARINGS	$p = \dfrac{50{,}000}{1 + \dfrac{(12\,L)^2}{18{,}000\,r^2}}$
CAST-IRON COLUMN SQUARE BEARINGS d	$p = \dfrac{80{,}000}{1 + \dfrac{(12\,L)^2}{800\,d^2}}$
CAST-IRON COLUMN SQUARE BEARINGS d	$p = \dfrac{80{,}000}{1 + \dfrac{(12\,L)^2}{1{,}067\,d^2}}$
WOOD COLUMN SQUARE BEARINGS b	$p = \dfrac{5{,}000}{1 + \dfrac{(12\,L)^2}{250\,b^2}}$

Machinery, N.Y.

Fig. 4. Strength of Columns, from Gordon's Formulas, adapted from Hodgkinson's.

Fig. 2 is a chart for the strength of flat plates, and most of the formulas are due to Grashof. Very little explanation is necessary, as the formulas are all rather simple. When the constant K is used, its value is found from $\frac{d}{D}$ or vice versa. The notation used in the different charts is given in the table on page 526.

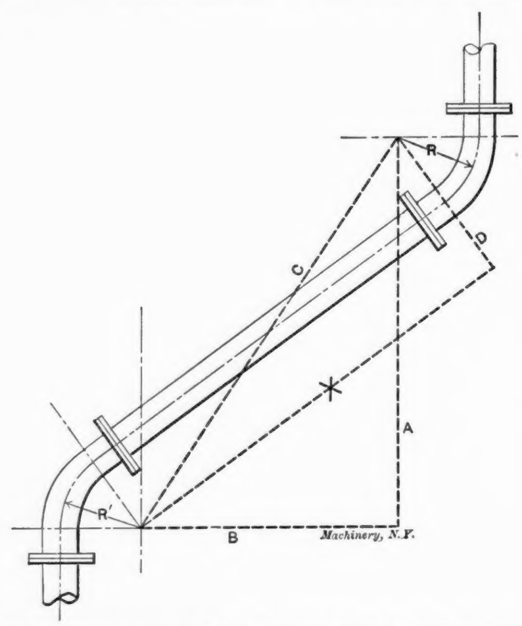
Having the formulas in tabulated form it only remains for the draftsman to select the correct one and apply it. The habit of correctly analyzing simple cases is easily acquired and is a long step toward the goal that every draftsman should strive to attain.

I might add in closing that the habit of checking all calculations by one's eye is a fine thing to cultivate, for sometimes an error will occur that is so ridiculously wrong that it is immediately "spotted" and the work revised to conform. Many times other questions than mere strength enter in, as for instance, rigidity as required by a machine tool; or strains due to expansion and contraction as shown in the cooling of cast iron, especially in the arms of pulleys and bevel gears. In such cases empirical formulas and experience are the best guides.

* * *

METHODS FOR OBTAINING THE DISTANCE BETWEEN ELLS TO CONNECT PARALLEL LINES OF PIPE.

Mr. E. C. Falk, Torrington, Conn., submits his method, illustrated below, for obtaining the distance between two ells to connect two parallel lines of pipe. Referring to the solution of this problem in the February number of MACHINERY by Mr. Franklin H. Smith, Mr. Falk states that he considers his solution simpler, and that it can be used without regard to the magnitude of the angles, and without the use of constants.



The Distance between Ells to Connect Parallel Lines of Pipe.

In the figure, A, B, R and R' are generally known or determined beforehand, B being the distance between the two ells of pipe minus the sum of the two radii, and A being the distance parallel to the lines of pipe between the points of offset of the two ells, which points can be determined from the maker's catalogue if standard ells are used. Then

$$C = \sqrt{A^2 + B^2},$$
$$D = R + R', \text{ and}$$
$$X = \sqrt{C^2 - D^2}, \text{ which makes the formula}$$
$$X = \sqrt{A^2 + B^2 - (R + R')^2}$$

If the distance between the flanges of the ells is desired, this will be the distance X minus the sum of the two offsets of the ells.

* * *

Under Shop Receipts and Formulas, April, 1905, the item "Steel Hardening and Tempering Compound" was attributed to H. S. Hindman, Columbus, Ohio. It should have been W. S. Hindman, Columbus, Ga.

REPAIRING FIELD COILS.

NORMAN G. MEADE.

Only a general description of field-coil winding, with a few illustrations, will be given in this article, as a specific description of the many styles would occupy too much space.

Fig. 1 gives a view of a form for winding coils that have no spools. The shape of this form depends on the style of the coil to be wound. The form is clamped to a lathe faceplate either by bolts extending through the form or through the side *b*. Dowel pins *c* and *c′* serve to hold the side *b′* from twisting, and the bolt *d* and the nut *e* hold the whole form together securely.

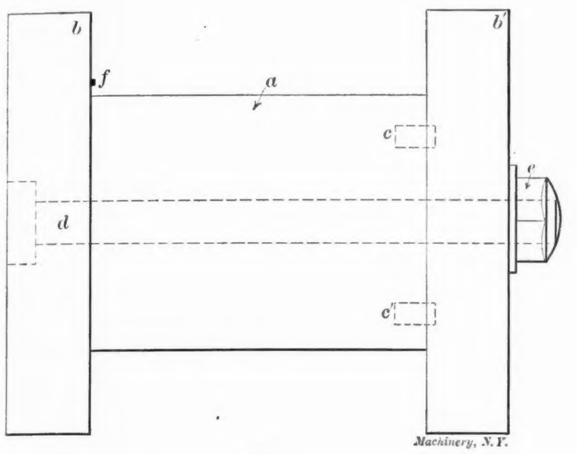


Fig. 1.

Fig. 2 represents a guide that is held in the toolpost of a lathe. Attached to the piece *a* is a grooved wheel, *b*, over which the wire from the reel runs. The same arrangement of reels can be employed in winding field coils that is used with armature coils.

Fig. 3 shows front and side views of a connector, which consists of a piece of sheet copper, *b*, rolled up at the end, *c*, and sweated into a sleeve, *a*, at the opposite end. The sleeve has a setscrew, *d*, for outside connections on the machine.

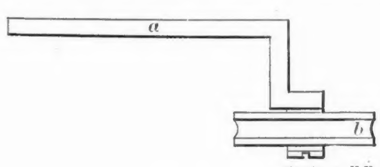


Fig. 2.

A very convenient way of securing the last turn of wire is shown in Fig. 4. Here *a*, *b*, *c*, *d* and *e* represent the convolutions of wire; *f* is a loop of cotton tape with its ends protruding at *g*. The loop is laid on the coil before the turns *c* and *d* are made, then the end of the wire, *h*, is pushed through, and the loop *f* is drawn tight by pulling on the ends at *g*.

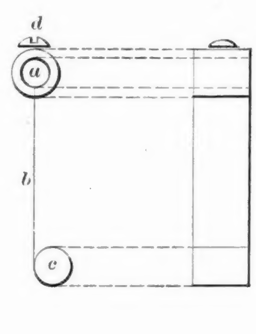


Fig. 3.

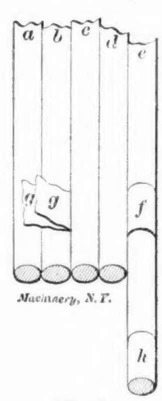


Fig. 4.

We will now start to wind a coil. First, bolt the form to the faceplate of the lathe, then arrange the reel conveniently for paying off the wire. Solder a connector like that shown in Fig. 3 to the end of the wire, and tape thoroughly. Catch the connector behind the pin *f*, provided for that purpose (Fig. 1), and proceed to wind. The wire running over a guide wheel on the toolpost enables the operator to use the tool carriage for guiding the wire as it is wound. With a little practice the operator will be able to run the tool carriage backward and forward with enough skill to permit considerable speed in winding.

The connectors can be made in different lengths and widths for varying styles of coils. The connector for the outer end of coil will, of course, be short, allowing the sleeve *a* to come on outside of insulation. Having wound the desired number of turns onto form, finish the end with a loop of tape, as shown in Fig. 4, and solder on outside connector.

Coils wound of wire fine enough to be flexible do not require connectors, as the wire itself may be left protruding through the covering. Before starting the winding, several pieces of cord or cotton tape must be laid across the form, with ends long enough to tie over the completed coil. Having completed the winding of coil, take off the side of form *b′*, and remove the coil.

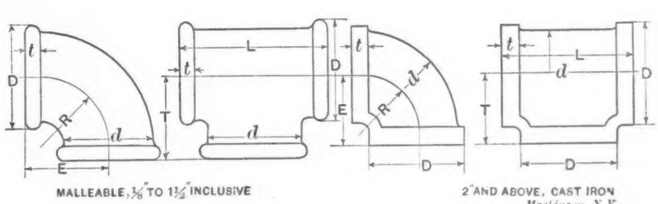
Different manufacturers have various methods of insulating their coils, and it is always well to treat the new coil in the same manner that the original coil was covered.

* * *

The impossible appears to have been accomplished in the maintenance of a clear fresh atmosphere in an office building in the heart of the Pittsburg mill district. This result has been secured in the case of the office building of the H. K. Porter Co., by the introduction of the special plenum heating system and coke air washing apparatus. The entire equipment was designed and installed by the B. F. Sturtevant Co., of Boston, through its local Pittsburg office. The heating apparatus consisting of fan, heater and belted motor, is installed in the basement in conjunction with the washer which consists of a metal supporting frame filled with broken coke, over which water is allowed to trickle. The air as it passes between the fragments of coke is thoroughly cleansed of smoke and dust, which is washed down by the water to the bottom of the device and there removed. Previously to the installation of this plant, drawings, papers and the like became very dirty. It is now reported that they are kept perfectly clean at all times, owing to the fact that no air can go into the building except through the heating apparatus, where it is very thoroughly and effectively cleansed. The slight pressure maintained within the building causes outward leakage at all points. It is anticipated that the same equipment will prove very advantageous for ventilating purposes during the summer time.

* * *

DIMENSIONS OF ELLS AND TEES FOR WROUGHT IRON PIPE.



Size.	*E*	*R*	*D*	*d*	*t*	*L*	*T*
1/8	5/8	9/16	13/16	9/16	3/16	1 1/8	5/8
1/4	3/4	5/8	1	3/4	3/16	1 1/2	3/4
3/8	7/8	3/4	1 1/8	7/8	1/4	1 3/4	7/8
1/2	1 1/8	7/8	1 1/4	1 1/16	1/4	2	1 1/8
3/4	1 3/8	1 1/16	1 9/16	1 5/16	5/16	2 1/2	1 3/8
1	1 9/16	1 1/4	1 7/8	1 5/8	5/16	2 7/8	1 9/16
1 1/4	1 7/8	1 1/2	2 1/4	2	5/16	3 1/4	1 7/8
1 1/2	2	1 5/8	2 1/2	2 1/4	3/8	3 5/8	2
2	2 3/8	2 1/8	3 3/8	2 7/8	3/4	4 3/4	2 3/8
2 1/2	2 7/8	2 1/2	4	3 1/2	3/4	5 5/8	2 7/8
3	3 3/8	2 3/4	4 5/8	4	7/8	6 3/8	3 3/8
3 1/2	3 5/8	3 1/8	5 1/4	4 5/8	7/8	7 1/8	3 5/8
4	4	3 3/8	5 7/8	5 1/4	1	8	4
4 1/2	4 1/4	3 3/4	6 3/8	5 7/8	1	8 5/8	4 3/8
5	4 1/2	4	7	6 3/4	1 1/8	9 1/2	4 3/4
6	5	4 5/8	8 1/4	7 5/8	1 1/8	11	5 1/2

Contributed by W. I. Mann, Pittsburg, Pa.

PROPORTIONS AND CHARACTER OF SAND AND GRAVEL TO USE IN CONCRETE.*

The terms gravel and sand are a trifle indefinite and obscure. Sand, the grains of which are above a certain size, is called gravel, and there is no distinct line of demarcation between these two, one gradually shading into the other. For convenience we will say that all of the mixture which will pass through a No. 10 sieve is sand, while that which will not shall be called gravel.

The relative proportions of sand and gravel in concrete can only be fixed after careful tests of the material to be used. The writer's experience has been that gravel taken from different sources varies so much in size, that the proportion of sand to be added to make a good concrete would have to be determined in each instance. It would, therefore, appear to be impossible to make a definite specification that would cover the proportion of sand and gravel taken from different localities. In order to obtain the correct proportions to be used, the material in question should be screened, separating gravel from the sand, and the proportions of voids determined in each.

Good authorities state that loose gravel of ¾-inch diameter and under contains about 33 1-3 per cent of voids, which is reduced to 21 per cent by ramming. Ordinary loose sand contains about 38 per cent of voids, which is reduced to 21 per cent by ramming.

From the above it can be seen that one part of sand will fill the voids in three parts of gravel of the size mentioned. That is a mixture of 1:3. The per cent of voids increases somewhat with an increase in the size of the gravel, which would correspondingly decrease the above ratio. The gravel should be of different sizes, varying from coarse to fine, screened or washed entirely free from clay, loam, or other foreign matter, and be free from scale, slime or humus.

Gravel sufficiently clean can often be obtained from river beds or gravel bars, but rarely in gravel banks. Gravel can readily be cleaned by sluicing. Sufficient water should be used to carry away all objectionable matter, including fine sand. By using a good sluicing plant, the writer has washed out an excellent quality of gravel from banks containing a large percentage of clay.

The sand should be clean and sharp—preferably coarse and fine mixed—and entirely free from all foreign matter. Several authorities cite laboratory experiments to prove that a percentage of loam is not objectionable. However, the writer would not favor using other than clean sand in actual work.

During the past three years we have used, approximately, the following proportions with satisfactory results: For concrete in arches and girders, cement 1 measure, sand 1½ measures, gravel 4½ measures. In piers and abutments, cement 1 measure, sand 2 measures, gravel 6 measures. In spandrels and retaining walls, cement 1 measure, sand 2 measures, gravel 5 measures. For copings and parapet walls, cement 1 measure, coarse and fine sand mixed 3 measures. The work has shown no surface cracks, or spider-webs. The writer's experience would indicate that more cement is required in gravel concrete than in same volume when broken stone is used. We believe that 10 per cent is about the amount of extra cement that should be used in gravel concrete over and above that required with broken stone.

There is no question in the writer's mind as to the sufficiency of gravel concrete. In every instance where we have used it, we have found it to compare favorably with concrete made of broken stone. The gravel concrete is denser, and requires less ramming. We have built several concrete-steel girder bridges, of spans varying in length from 8 to 25 feet. In these we have used gravel and sand concrete, with gratifying results. The mixtures have been as specified above, and we see no good reason why properly proportioned gravel and sand concrete could not be used in constructing arches of any length desired.

* * *

The pins used in the chord system of the new Blackwell's Island Bridge, New York, are nickel steel, 16 inches diameter and 9 feet ⅝ inch long. Each pin weighs 6,050 pounds, and 300 are required in the structure.

* J. H. Marsh, *The Iowa Engineer*, March, 1905.

PRACTICAL DONT'S FOR MACHINISTS.

H. E. WOOD.

Don't say "that's good enough."
Don't turn a reamer backwards.
Don't use a file for a pinch bar.
Don't grind a drill out of center.
Don't borrow tools; buy your own.
Don't try to cover up your mistakes.
Don't let your lathe run and cut air.
Don't be always looking for pay day.
Don't set a lathe tool behind the chip.
Don't file against the scale of cast iron.
Don't grind ditches in the emerywheel.
Don't use a monkeywrench for a hammer.
Don't use a screwdriver for a cold chisel.
Don't be always looking for quitting time.
Don't try to make a finishing cut on scale.
Don't drive a key too hard while fitting it.
Don't use a file without a handle on a lathe.
Don't run any machine with dull tools in it.
Don't run a machine with the belt too loose.
Don't use any more waste than is necessary.
Don't waste time by doing unnecessary work.
Don't grind round corners on the emerywheel.
Don't be too important to do insignificant jobs.
Don't run the point of your drill into the table.
Don't use the tail end of your vise for an anvil.
Don't always have a litter and muss around you.
Don't take off your overalls before quitting time.
Don't use the ways of a lathe for a bench block.
Don't slide rough castings along on a planer bed.
Don't believe that any two men will caliper alike.
Don't make a practice of being late to your work.
Don't cut a bevel gear without taking a central cut.
Don't do "government work" on the company's time.
Don't try to fool your foreman, for you may get left.
Don't use your employer's oil to wash your hands in.
Don't wait until Monday morning to fill your oil can.
Don't stamp steel letters or figures into cast iron scale.
Don't run the point of your lathe tool into the mandrel.
Don't rap the chips out of your file on the lathe shears.
Don't deny spoiling a piece of work if you have done it.
Don't watch for a chance to wash up before quitting time.
Don't look at the head of a chisel when you are chipping.
Don't set a lathe tool below the center for external work.
Don't do any unnecessary hammering on a drill press table.
Don't be too eager to make an impression on your employer.
Don't keep one eye on the boss and the other on your work.
Don't borrow bolts, straps, or clamps, and not return them.
Don't think yourself above asking questions for information.

Don't think that you know more than your foreman, even if you do.

Don't engage in extended conversation during working hours.

Don't be "woolgathering" when you are receiving instructions.

Don't gage the grip on your hammer handle to suit your pay.

Don't bear down on a hacksaw or file on the backward stroke.

Don't use the end of your hammer handle for a driving block.

Don't screw bolts and nuts hard enough to strip their threads.

Don't start a machine without knowing that everything is ready.

Don't screw a bolt into a newly tapped hole without oiling it.

Don't put milling cutters, taps, dies, or any tools away dirty.

Don't use a vise more than a month without oiling the screw collar.

Don't use one mandrel to drive another one out of a piece of work.

Don't wear jewelry on your hands in a machine shop; it is dangerous.

Don't do any hammering on any machine if it can possibly be avoided.

Don't file a piece of work when you can make it to size with a tool.

Don't try to drill a piece of work unless it is properly supported.

Don't throw files, one on top of another into a drawer, or on the bench.

Don't drill a hole unless you know that the drill is properly ground.

Don't hold a tool of any kind on an emery wheel until it gets blue.

Don't fit up a key without trying a straightedge on it occasionally.

Don't ask your neighbor for instructions; go and ask your foreman.

Don't leave your tools laying around after you are through using them.

Don't ream out holes in aluminum without kerosene oil on the reamer.

Don't say that you understand your instructions unless you actually do.

Don't make a piece of work too small and then bend the gage to fit it.

Don't start up a lathe without seeing that the tailstock spindle is locked.

Don't tap a hole and put a bolt in it without cleaning out the chips.

Don't put an arbor or shaft on lathe centers without lubricant on them.

Don't grind on the flat side of an emery wheel; the edge is made to grind on.

Don't ream out a hole in any kind of steel or wrought iron with common machine oil.

Don't work to a caliper that has been set by another man; set it yourself.

Don't start to make a piece unless you first know what you want to make.

Don't leave a machine after running it, without cleaning it up after yourself.

Don't use a new file on rough castings, as a partly worn out one is much better.

Don't true up a flywheel or gear blank to the surface that you are to turn off.

Don't leave too much stock on a piece of work to take off with the finishing cut.

Don't be jealous of your fellow workman if he gets a better class of work than you do.

Don't try a steel gage or an expensive caliper on a shaft while it is running.

Don't grind a lathe tool for cutting brass, the same as you would for cast iron.

Don't cut the teeth in a gear blank unless you know the outside diameter is correct.

Don't think that you are the only one that has troubles; the foreman has his also.

Don't put a mandrel into a newly bored hole without a lubricant of some kind on it.

Don't use any kind of oil but kerosene to start a shaft loose that has begun to cut.

Don't let the point of your drill touch your work and run along without feed on it.

Don't do a job and leave it without cleaning off all the burrs and marks made by yourself.

Don't put a piece of work on centers unless you know that the internal centers are clean.

Don't turn a straightedge cornerwise when testing a piece of work, as it will fool you.

Don't run a machine when any part of it is out of order, without notifying your foreman.

Don't think that you are a machinist just because you have walked under a sign that read "Machine Shop."

Don't consume any more than your time allowance in cleaning up your machine on Saturdays.

Don't bother your foreman with *all* your little troubles, but try to fight them out yourself.

Don't always have some different way to do your work than the way the foreman tells you to do.

Don't think that a new foreman is not a mechanic because you never saw him do the work.

Don't run a drill an instant after it begins to squeal, but take it out and ascertain the cause.

Don't start up a planer, shaper, or milling machine unless your work is securely fastened down.

Don't try to straighten a shaft on lathe centers, and expect that the centers will run true afterwards.

Don't put a piece of work on lathe centers unless you know that all your centers are at the same angles.

Don't believe that a V-thread will lift as big a load as a square one will, when all other things are equal.

Don't take it for granted that another man's measurement is all right, but go and measure for yourself.

Don't try to take a burr or bruise off from a flat surface with a scraper or the tip end of a file.

Don't think because you have always done a piece of work in one way, that there is no other way to do it.

Don't think because a piece of work has been done in one way for twenty years that it is the correct way.

Don't rub a surface-plate on your work too much at one time; it is bad for both the plate and work.

Don't put enough lead on a surface plate to make it smear when you try it on the surface you are scraping.

Don't take a difficult piece of machinery apart without having each piece marked to show where it goes back.

Don't set the cutting point of a lathe or planer tool any farther out from the tool rest than is absolutely necessary.

Don't stop with an oil channel until you get clear through and out with it, so as to give the air a chance to escape.

Don't put a nice piece of finished work in a vise without using a set of false jaws made from some soft material.

Don't put collars on a milling machine mandrel without being absolutely sure that there is no dirt or chips between them.

Don't touch a piece of work with a file or scraper unless you know what you want to do to it, and what you are doing it for.

Don't finish your job and ask your foreman for another one without putting your tools away and cleaning up the muss you have made.

Don't take a lathe center out of its socket without having a witness mark on it, and put it back again according to the mark.

Don't start up any machine unless you are absolutely sure that your cutting tools are screwed up tight enough so they will not slip.

Don't start polishing a shaft on lathe centers without having it loose enough to allow for the expansion by heat from the polishing process.

Don't stop the feed on a drill in the middle of a piece of work and let the drill run along without backing it off away from the cutting surface.

Don't put any two pieces of machinery together without making absolutely sure that there is no dirt, or chips, or bruises, or anything to prevent the surfaces from coming promptly together.

* * *

A new use of acetylene gas as an explosive is described in a recent report of Consul-General Guenther, Frankfort, Germany. He says that for this purpose carbide of calcium is reduced to small particles and put into a cartridge, consisting of a tin box. In this the carbide lies at the bottom and above it is a partition filled with water. Above this is a vacant space with the electric percussion device. On the side of the cartridge is an iron pin by means of which the partition between the carbide and the water can be perforated. After the drill hole has been completed the cartridge is placed in it and the hole is closed with a wooden stopper. Then the protruding iron pin is dealt a blow, by which the partition is perforated and the water is caused to come in contact with the carbide, whereby acetylene gas is generated. This mixes with the air of the drill hole. After five minutes the gas is ignited by an electric spark. By this method of blasting the rock is said to be not thrown out but rent with innumerable cracks, so that it can be easily removed afterward.

Copyright, 1905, by THE INDUSTRIAL PRESS.

Entered at the Post-Office in New York City as Second-Class Mail Matter.

MACHINERY

DESIGN—CONSTRUCTION—OPERATION.

PUBLISHED MONTHLY BY

THE INDUSTRIAL PRESS,

66-70 WEST BROADWAY, NEW YORK CITY.

LESTER G. FRENCH, Editor.
FRED E. ROGERS, Associate Editor.

The receipt of a subscription is acknowledged by sending the current issue. Remittances should be made to THE INDUSTRIAL PRESS, and not to the Editors. Money enclosed in letters is at the risk of the sender. Changes of address must reach us by the 15th to take effect on the following month; give old address as well as new. Single copies can be obtained through any newsdealer.

We solicit communications from practical men on subjects pertaining to machinery, for which the necessary illustrations will be made at our expense. All copy must reach us by the 5th of the month preceding publication.

JUNE, 1905.

PAID CIRCULATION FOR MAY, 1905,—21,806 COPIES
Including 600 advertisers' copies, as part of their contracts.

MACHINERY is published in four editions. The practical work of the shop is thoroughly covered in the Shop Edition—$1.00 a year, comprising more than 430 reading pages. The Engineering Edition—$2.00 a year—contains all the matter in the Shop Edition and about 250 pages a year of additional matter, which includes a complete review of mechanical literature, and forty-eight 6 x 9 data sheets filled with condensed data on machine design, engineering practice and shop work. The Foreign Edition, $3.00 a year, comprises the same matter as the Engineering. RAILWAY MACHINERY, $2.00 a year, is a special edition, including a variety of matter for railway shop work—same size as Engineering and same number of data sheets.

A reader of MACHINERY has written the editors a friendly criticism: "The radius lathe tool described by one of your correspondents is not new," he says, "for we have used one since Noah made a floating machine shop of the ark." No, the tool is not new, and we doubt whether in the last analysis one in ten of the devices shown month by month in MACHINERY is new, in spite of the fact that we have among our contributors some of the brightest and best mechanics to be found in the machine shops of the country. It is not easy to hit upon things that are strictly new, and in fact, the old and tried devices are usually the best. A good description of such a device with full directions for its design, construction and use may be of more value than a whole paperful of kinks that have not proved their worth. There is a crop of young mechanics coming along each year who need to learn about the old things as well as the new, and however old and familiar a device may be, it is more than likely to be new and interesting to at least a few of the thousands of readers of this publication. Most machinists know all about radius tools to which our friend refers, and have used them; but we believe when a reader has such a tool to design, he will be glad to remember that by turning to a certain page of MACHINERY he will find all the information necessary to enable him to make one, and thus save himself a lot of trouble and time in designing and experimenting.

* * *

THE CRUCIAL TEST OF THE TURBINE.

The steam turbine has proved its worth and made a place for itself and the manufacturers of turbines have not hesitated to acquaint the public with its many good points and advantages. The defects of the turbine, however, are not so easily to be found out. It is well known that things have happened in starting up and running turbines in various parts of the country that the manufacturers would just as soon not advertise in bold-faced type and would be glad to forget if they could. We believe, however, that most of these difficulties are no more than must be expected in establishing any new enterprise and that they are mostly defects which can be overcome. Just which are of this character and which are inherent in the turbine itself can be told only after years of service under varied conditions.

The vital question, it seems to us, is whether the turbine will prove durable. We know what it is capable of doing under shop test, but how about the economy of a turbine which has been in constant operation ten or twenty years? In what condition will the blades be and how many times will the average turbine have to be rebladed during such a period? This we believe to be the crucial test of the turbine, which will determine whether the turbine or reciprocating engine is to hold sway in the power plants of the future.

Some light was thrown on the subject last month by photographs supplied by the Westinghouse Machine Company, showing the present condition of the interior parts of a turbine installed at the Wilmerding plant some five years ago, where it has been in operation 24 hours a day ever since. Blades which have been subjected to the action of steam during this entire period, and most of the time under unfavorable conditions, show practically no erosion or cutting action and indicate that turbines under conditions existing at this plant, at least, are good for a long period of service without deterioration. If this should prove to be the case in other plants as well as this one, an important point will have been demonstrated, viz., that the turbine is an engine which will run with nearly or quite as good economy when old as when new. The steam engine will not do this without occasional repairs, such as reboring the cylinder, refitting the valves, etc., and is furthermore liable to use steam extravagantly through faulty valve setting. More information upon the erosion of turbine blades is needed and needed badly.

* * *

SHOPWORK INSTRUCTION.

The article in this number upon the very successful system of shop training that has been developed at the Worcester (Mass.) Polytechnic Institute will be read with interest by many. A fundamental principle in the instruction of young men at this institute is that comprehensive shop training is desirable and necessary in connection with engineering studies. More time and attention are here given to the subject of shopwork, probably, than at any similar institution. If this meant that more time were spent in teaching the boys to turn out good-looking and accurate reamers, gears, surface plates and whatnot, than is usually given to such work in shop-work courses, and nothing else, we should say the time could be used to better advantage.

But this is not the case. The effort is made to rise above the mere turning out of work of a given quality, since to become dextrous with tools or drawing pen, to the neglect of broader work not so easily learned, is not what tuition fees are paid for. At Worcester they have the student grapple with the commercial side of shopwork, which, it may be added, is also the engineering side. He learns that the time element is just as important as fine finish or accuracy; that certain methods are better than others because they save time; that high-speed steels and heavy machines are useful because they also save time. Just how much time can be saved is a subject for study and requires estimating, planning and laying out work, and extended experiments and investigation, all of which lead at once to studies as important for the student of engineering as any laboratory work that he may be called upon to do. It is just as necessary for an engineer to understand how money can be saved by saving time in the shop, as to know how money can be saved at the coal pile by making certain adjustments in the valve gear of a Corliss engine. One subject is just as scientific as the other, and has just as much to do with engineering, although the shop-work problems have not generally had the same attention as have the so-called laboratory exercises. Why, we do not know, unless supposed to be less scientific.

* * *

The September number of MACHINERY is to be a draftsmen's number. There will be nothing "special" about this number, either in appearance or size, but the editors have on hand, or in prospect, several contributions of unusual value to draftsmen, and these will appear in September, making that an issue which many draftsmen will want to keep for reference. It is probable that some who read this will be glad to assist us in the preparation of this number, and we will ask any who have information or data that would be appreciated by draftsmen to submit the material for our inspection. The contributions must be short, however, and should be marked "For the draftsmen's number."

ENGINEERING REVIEW.

CURRENT EVENTS, TECHNICAL AND MECHANICAL—LEADING ARTICLES OF THE MONTH.

The metal calcium has successfully gone through an experimental stage of production similar to that recently passed through by aluminum, and is now being commercially reduced by electrolysis, the process being similar to that now successfully used for aluminum. In this process, as carried on by a German firm, the molten metal is formed at the cathode terminal of an electrolytic chamber. It is thought that the metallic calcium will serve many useful purposes, there being a large demand for the metal in connection with hardened steel. An English contemporary states that the price of calcium per ounce in 1903 was £56, and the same quantity can now be purchased for 1s. 6d., with prospects for further reduction. This great reduction is caused by the success of the new process.

In an article describing the erection of the 5,000-K.W. engine-driven alternators of the New York Subway, by Mr. R. L. Wilson in the *Engineering News*, it is stated that the central portion of the revolving field element is a steel casting weighing 20 tons. It has a bore 37 inches in diameter with a "stepped" fit on the shaft, each step being 8 inches long; a 10-inch space is cored out in the center, making the total length of the hub bearing 42 inches. The allowance for pressure was 0.008 inch but in practice this was found to be slightly excessive, and it was reduced to 0.0065 to 0.007 inch. With the latter allowance the pressure required to force the steel hubs into position on the shafts was from 500 to 600 tons. The shafts are hollow nickel steel forgings, each having a hole 16 inches in diameter running the full length.

Diamond drill borings of great depth have been made on the gold-bearing reefs of the Transvaal. By this method of boring a solid core of rock is removed, in sections, of course, as the boring proceeds, and in this way a fairly accurate knowledge of the rock formation can be obtained throughout the depth of the bore without going to the expense of sinking a shaft. According to the *Mining Reporter* the deepest diamond drill borings are near Johannesburg and Doornkloof, these bores being 5,582 and 5,560 feet deep respectively. A core of 1⅜ inch diameter was removed for the greater part of the distance in each case. The Johannesburg hole required nine months for sinking and that at Doornkloof about fourteen months. An interesting feature of this work from our point of view is that American machinery is almost exclusively used for it in the South African field.

The growing use of electricity for heavy railway operations was dwelt upon by Mr. George Westinghouse in an address given during the opening exercises of the American Railway Appliance exhibition held in Washington in connection with the International Railway Congress. He said: "A new era in railway operations has dawned with its many new problems. I refer to the growing use of electricity for the movement of trains. There have already been such demonstrations of the benefits to be derived from the substitution of the electric motor for the steam locomotive that it requires no great prophet to predict the extensive growth of electric traction upon the great railways of the world and the eventual replacement of the steam locomotive. Fortunately, the time element, which is such a controller of events, and the financial problems involved, will ensure gradual development and extension of the use of electricity."

It is said that the municipal authorities of Vienna, Austria, have decided to equip the city's professional fire brigade and the auxiliary volunteer fire brigade entirely with self-propelled vehicles, 53 motor chemical engines having already been ordered to replace horse chemical engines. The cost of the change will be in the neighborhood of $180,000, but it is said it will effect an annual saving of over $15,000. It would seem as though the motor-driven fire engine would be much more efficient in getting to fires than the horse machine and hence should be bound to come into general use. The engine would naturally be propelled by steam if the steam pump is to be retained, but much may be said in favor of an electric machine having a battery equipment and electrically driven pumps. The battery equipment would not need to be great, since the distances covered would be relatively small. The supply for operating the motors would be drawn from stations which might be located at each fire hydrant.

In a paper read by Mr. D. B. Morison before the North East Coast Institution of Engineers, attention was particularly directed to the danger of oil in boiler furnaces and to the fact that collapses of furnaces are almost always the direct result of scale or of oil in the feed water, particularly the latter. Mr. Morison said that no ordinary furnace fails from lack of strength if clean and covered with clean water; failure is the result of scale or dirty water, and above all of oil, and a very thin smear of it has an effect totally out of proportion to what might be expected. Starting out with a furnace having a factor of safety of five, it is found that this factor rapidly decreases when the temperature exceeds 650 degrees F., vanishing entirely at a red heat. Steam at a pressure of 200 pounds per square inch, for instance, has a temperature of 380 degrees F., and, as stated above, the temperature of 650 degrees F. marks the commencement of loss of tenacity of the steel composing the furnace walls; at 1,200 degrees F. about 75 per cent. of the tenacity of the steel is gone. A clean furnace rubbed over with a very clean coat of mineral oil will at once rise in temperature to over 650 degrees even under a light duty, hence the danger of even very small quantities of oil or grease in boilers. High grade mineral oils do not give near the trouble in marine boiler practice as do low grade oils, which emulsify, and therefore cannot be filtered out of the feed water without chemical treatment.

The argument formerly advanced against the use of forced draft instead of induced draft, that it burns out the grates, seriously injures the boilers, and blows gas and smoke from the fire doors, is now seldom heard. The basis of this opinion originated in the experience of some engineers with plants equipped with fans operated at far above the proper speed. This was the result of installing (through ignorance) a fan too small for the work, and then forcing it above its normal speed in order to secure the desired air volume. As a consequence, instead of creating an ash pit pressure of ¾ to 1¼ inches, which is all that is ordinarily required, the pressure was forced up to even 5 or 10 inches, with the attending objectionable results.

In one instance, the engineer complained of gas discharged from the fire doors with incidental effects, and condemned forced draft in toto, although he was favorably disposed toward induced mechanical draft. Investigation showed that the fan was being operated at about 12 inches water pressure, which at once accounted for all the trouble.

When forced draft is used, the air as it passes from the ash pits to the combustion chamber is greatly reduced in pressure, owing to the resistances of the grates and the fuel. Coincidentally, the stack, even if a short one, tends to produce a partial vacuum in the furnace. As a result it is practically impossible to create under proper conditions more than a slight excess of pressure in the combustion chamber, and this should not be enough to force the gases out at the fire doors.

Accurate knowledge regarding the proper application of the fan blower for this purpose, will readily dissipate any false impressions regarding forced draft.

A new and improved type of storage battery has been developed by Mr. Joseph Bijur, who had as collaborator, Dr. J.

S. C. Wells, of Columbia University. This cell is of the old lead and sulphuric acid type, but one having the plate differently constructed from the old form, the end sought in the construction of this cell being the attainment of a rigid structure, which should be free from any tendency to buckle and which should, at the same time, allow free circulation of the acid and hold the active material so that it could not be displaced. The plate as constructed, consists of a number of small grills which are supported at two opposite ends by being firmly welded to the supporting grid. These grills are cast in molds under pressure, and so are one piece of lead, but for purposes of description they may be compared to a number of flat ribbons of lead, held together by short intermediate cross pieces. The grills so formed are fastened at intermediate points to the terminals, which are welded to the supporting grid, this method of support allowing the grill to expand lengthwise freely.

With this plate, formation chiefly takes place in the rectangular openings of the grill, a large surface area being exposed to the electrolyte, since these openings are small. As these openings are never completely closed by the active material, the acid has access to the interior of the plate. The active material, as formed, wedges itself into the rectangular openings of the grills, expanding the latter into an elliptical shape, and is thus held tightly in place against the lead of the grill. This formation exposes a large surface of the active material which, being always in contact with the electrolyte, is said to permit high rates of charge and discharge without injury to the plate, and with but little polarization. The grills are welded solidly to the grid without the use of solder.

Railway men in the United States are just waking up to the possibilities of other methods of transportation besides the steam locomotive and the electric trolley car. In many parts of the East and West high-speed electric lines have been built parallel to existing steam roads and by reason of more frequent service at lower fares, they have cut into the business of the steam road to such an extent that the running of trains is attended with continual loss. But, on the other hand, the equipment of a high-speed electric railway is very costly and it can only be done where the volume of traffic is considerable. Between the steam railway and the electric railway lies a field for the steam or gasoline motor car. Light railways equipped with motor cars have been in successful operation for some years in Europe, and experiments are now being conducted on the Union Pacific and other railways in this country with a view of developing what might be said to be a lifesaver for the old steam roads.. A steam road can put on motor cars and undoubtedly successfully compete for passenger business with the electric lines which have cut so seriously into their business. At the recent meeting of the International Railway Congress at Washington, the subject was touched upon in the following language: "It may be expected that from now on automobile cars and auto motors hauling trailers will constitute a valuable means of transportation, which on some lines will have a great future. Owing to the saving in the number of employes required, the probable reduction in cost of maintenance, the material reduction in the cost of traction and better utilization of rolling stock, and the smaller extent of station installations required, it will be possible materially to reduce the cost of working lines with little traffic, and will, in the case of other lines, result in a material improvement in the working of some classes of service. Their use will certainly effect a change in the system of operation in the case of a great number of lines, and appears to have a real future before it."

THE GAYLEY DRY AIR BLAST SYSTEM.

Indian and Eastern Engineer, April, 1905.

A very important departure has been made at one of the blast furnaces of the Carnegie Company at Pittsburg, by the addition of a cold storage plant to the blast. Makers of pig-iron are familiar with the fact that it is very difficult to control the output of pig, with changing temperatures and changing weather. Sudden cooling of the metal has been frequently observed, due to no apparent cause, and this has led blast furnace managers to carry higher temperatures than would otherwise be necessary, to meet these sudden calls. Mr. Gayley, the manager of the works in question, has traced the trouble to the varying quantity of watery vapor present in the air that is pumped into the furnace. Records he obtained showed that there was a difference of from 2.18 grains of moisture per cubic foot of air in January to 5.6 in July, and 5.68 in September, and he calculated that this meant that 87 gallons of water were entering the furnace per minute in January, while in July and September over 200 gallons were entering in the same time. And this was not the whole of the trouble. The quantity of watery vapor present varied from hour to hour on the same day by changes of wind, for instance, and from other causes. On the same day the quantity of moisture present in a cubic foot of air varied from 1.96 grains at 8 A. M. to 3.06 at 8 P. M. The trouble has been overcome by drying the air, before it is allowed to enter the hot blast stoves. It will be remembered that combustion is maintained in a blast furnace by the continued pumping of air into the furnace, by a blowing engine, and that the air is heated by a portion of the waste gases from the furnace. An ammonia compression plant has been erected in the neighborhood of the blast engine, the ammonia expansion coils being made to cool brine in a tank, and the brine is taken to a cooling chamber, where it is caused to circulate through pipes arranged for the purpose, and around which the air is made to pass on its way to the hot blast stoves. The air, being cooled, loses a large portion of its ability to carry moisture, and the watery vapor it has carried is deposited upon the cold brine pipes, the air passing on cooled and dried. The moisture freezes upon the brine pipes, and it is arranged to disconnect a portion of them periodically, and to thaw out that section by passing hot brine through the pipes. The water produced is used for the boilers. The drying of the air not only gets rid of the uncertainty mentioned above, but it also adds to the value of the blast. Air which is cooled, as is well known, occupies less space than air which is warm, and as it is oxygen that is wanted in the furnace, and as the air is dealt with in the blowing engine by cubic feet entering its cylinder, the result of the cooling is that a larger quantity of free air, as it is termed, passes into the furnace, and a larger quantity of oxygen. The power required from the blowing engine has been reduced in consequence sufficiently to provide power to drive the ammonia compressor, and in addition the output of the furnace has been raised from an average of 358 tons per day to an average of 447 tons. It must be remembered that the output of the American blast furnaces is larger than that of those in the old country. In fact, the increased output due to drying the air is nearly equal to the output of many furnaces in the United Kingdom. The product of the furnace is also stated to be more uniform and therefore of higher value. There is one point in connection with this that is open to criticism. The arrangement for cooling the air is not the best, nor the latest. Cooling the air is largely practised in a modern refrigerating plant, but the action is made quite continuous. It would never do to have to lay off part of the apparatus to clear off the ice. The usual plan now adopted is to drive the air between a battery of galvanized iron plates, over which the cooled brine is constantly trickling.

NIAGARA WATER POWER.

William C. Unwin, F. R. S., in the Engineering Supplement, London Times.

At the present time, when in this country large schemes for the distribution of power are being carried out or are being projected, it may be opportune to give some account of the progress of power distribution at Niagara.

If all the water flowing down the Niagara Gorge and all the fall could be utilized, then in average conditions about 7,000,000 horse power would be available. But, though the outflow from the great lakes is singularly uniform, there is a variation in flow, and part of the fall must be wasted to obviate engineering works of too costly a character. The value of such a source of energy was perceived long ago. In 1861 a small canal was constructed, 35 feet wide by 8 feet deep, from above the upper cataracts to the top of the bluff

below the falls. At the edge of the bluff mills were erected with turbines driven by water from the canal. Later the State reservations were formed on both sides of the river to preserve the natural beauties of the falls, and this to some extent hindered efforts to increase the water power. In 1886 Mr. Evershed obtained a charter for utilizing power about a mile above the falls, the tail water being carried away by a tunnel and discharged inconspicuously into the lower river. But it was not till the discovery of electrical methods of distribution that Mr. Evershed's scheme became commercially practicable.

In 1890, the Cataract Construction Company was formed, with the object of developing 100,000 horsepower and with the right to utilize 100,000 horsepower more. though, at that time, it required great boldness and faith in the progress of electrical methods to expect that such a block of power could be distributed, over a sufficient area, without too great enhancement of cost. However, the two power houses of what is now the Niagara Falls Power Company on the American side have been completed, 105,000 horsepower of turbines are installed, and about 75,000 horsepower is regularly delivered partly near Niagara Falls, partly at Buffalo, 22½ miles distant, and at Tonowanda and Lockport.

An essential condition of the success achieved by the Niagara Power Company was the adoption, after prolonged discussion, of the system of distributing two-phase current with low periodicity, for this system lends itself to the most varied applications. In the two power houses the generating units are all identical, and any unit can be put on any service, the current from all the generators being two-phase current, at 2,200 volts, with a periodicity of 25. With such a current the voltage can be changed in the easiest and cheapest way by static transformers, single-phase current can be supplied if required, and direct current may be obtained without much increase of costly rotary converters. Thus, from the Niagara Power Company's station direct current is supplied for lighting; low tension direct current for electrolytic work, as for making caustic soda, bleaching powder, sodium; direct current for trolley tramways; and single-phase current for heating in electro-metallurgical processes. When the transmission to Buffalo was considered, it was found that there would be a saving of copper with a three-phase current, and a method was found of transforming the two-phase current to three-phase. The transmission to Buffalo is three-phase current at 22,000 volts. It is not generally realized in this country, how unsuitable the electric systems adopted in lighting stations are for a varied application of current for lighting, power, electro-chemical processes, and long distance transmission. The system at Niagara satisfies all requirements. The cost of transformation when it can be done on a large scale is very moderate.

Very early the Niagara Power Company obtained rights on the Canadian side, but action was postponed till experience had been gained with the plant on the American side. Now a power house is in course of construction immediately above the falls on the Canadian side, with a short tail-race tunnel. This power house is to have 11 units of 12,500 horsepower each, of which five are already installed. Three-phase current at 11,000 volts and 25 periods is generated, and connection is to be made to the power house on the other side, so that they may be able to help each other if necessary. Arrangements are being made to transmit 20,000 horsepower, probably at 60,000 volts, to Toronto, a distance of 85 miles, pending the completion of the works next to be described.

A second installation of 125,000 horsepower on the Canadian side is being constructed by the Toronto and Niagara Power Company. The current generated will be 12,000 volts, with 25 periods. A remarkable feature of this plant is that the tail-race tunnel passes right under the upper cataracts and discharges under the center of the Horseshoe Falls. The head works are remarkable for the boldness with which a cribwork dam of great length was built in the upper cataracts. The plans of the Niagara Power Company and the Toronto and Niagara Company are, from the engineering point of view, very similar.

A third company, the Ontario Power Company, is also constructing works for utilizing 180,000 horsepower on the Canadian side. Their plans are different. The water is taken from a point above the intakes of the other companies, and conveyed in steel pipes 18 feet in diameter, to the edge of the cliff below the falls, thence the water is dropped in 9-foot pipes to a power house in the gorge below the falls. One of the large pipes supplying water for 60,000 horsepower is already constructed. Here also three-phase current of 12,000 volts and 25 periods will be generated.

Meanwhile the owners of the old hydraulic canal on the American side have been stimulated to increase the development of power on somewhat analogous lines to those they first adopted. The old canal has been successively enlarged. In 1881 they constructed a station for 1,500 horsepower, and in 1896 a station for 34,000 horsepower. They are now carrying out an installation for 100,000 horsepower.

The total utilization of power, therefore, now projected amounts to 650,000 horsepower. The whole of the machinery for this development may not be erected for some time, but that great confidence is felt that it will be required may be inferred from the fact that the very costly headworks, wheel pits, and tail races are being projected for the full projected amount of power. The cost at which electricity is supplied is believed to be about £3 10s. per horsepower at Niagara and £7 10s. at Buffalo. This is for power supplied, if required, for 24 hours. At Buffalo the cost is probably not very much below that of steam power to ordinary large users who do not require 24-hour power. But it is in many respects more convenient than steam power.

The mean flow of the Niagara River is about 222,000 cubic feet per second. Suppose, what is about true, that 150,000 horsepower are now daily utilized, that the mean available fall is 160 feet, and that the efficiency of the turbines is 0.75. Then the daily demand for water is 11,017 cubic feet per second, which is 5 per cent of the mean flow, or 6¾ per cent of the *minimum* flow. But if 650,000 horsepower are utilized the demand for water will be 47,740 cubic feet per second, or 21½ per cent of the mean flow and 30 per cent of the *minimum* flow. Obviously, if no alteration of the appearance of the falls is at present perceptible, the alteration is likely in the future to be very considerable, especially as the depth of water over the American fall is very small.

DESIGN AND OPERATION OF THE SUCTION GAS PRODUCER.

R. Mathot, Engineering Magazine, May, 1905.

The early gas engines were designed for the use of illuminating gas, but the high cost of this fuel has led to the designing of various forms of apparatus for the production of a cheaper fuel, in which the various types of gas producers have attained considerable success. The first type of producer built, the pressure gas producer, required a separate steam boiler to operate the steam jet for delivering steam below the grate, this being done for the purpose of enriching the gas with the dissociated hydrogen of the steam as it passes through the incandescent coal bed. Also, as there is pressure beneath the grate it is necessary to have some form of water seal or tight joint at the bottom while the entire apparatus has to be made gas tight. A gas holder is also required since the production of gas is independent of the operation of the gas engine to which it is supplied.

In modern gas engines of both the 4-cycle and the 2-cycle types, one of the outward strokes is a suction stroke, this suction being employed to draw in the mixed charge of gas and air. In the suction gas producer this suction stroke of the engine draws the air through the coal bed instead of having it impelled by pressure. This substitution of suction for pressure not only dispenses with the separate blowing jet, but also causes a pressure slightly below the atmosphere to be maintained in the whole apparatus, so that there can be no leaks and there is no necessity for sealing the ashpit. The separately fired boiler, also, is replaced by what is termed a vaporizer. This consists of a vessel containing water raised by the heat of the gas nearly up to the boiling point, the vapor thus formed being delivered under the grate and sucked up through the coal bed with the air by the engine. The supply is automatically controlled by the demand in this producer, the produc-

tion of gas ceasing when the engine is stopped, and increasing when the speed of the engine is increased; hence there is no need of a gas holder. The general arrangement of the suction producer plant is given by Fig. 2, in which *A* is the producer, *B* the evaporator, *C* the scrubber, *D* the pressure equalizer, and *E* the engine. A sectional view of a suction producer is given in Fig. 1.

At first, that is four or five years ago, the designers of such apparatus tended to construct producers of too large dimensions. The result of this practice was to produce an abundant volume of gas; but this was of a poor quality because the temperature of combustion was kept too low. It must be remembered that the hydrogen, resulting from the dissociation of the water contained in a state of vapor in the air fed into the furnace, is the chief element which enriches the gas. The supply of air is a function of the dimensions of the machine, while the velocity of the air current increases as the area of the grate is reduced. The dissociation of the greatest amount of water is only made possible by a considerable velocity of the draft passing through the fire. It has been demonstrated that the greatest production of hydrogen and most effective reduction of dioxide to monoxide is with a producer, the cross section of the base of which varies between 0.6 and 0.9 times the area of the piston of the engine producing the suction. This applies to the simple 4-cycle engine running at a piston speed of 600 to 800 feet per minute. The depth of combustible in the upper part of the producer retort should be 2 to 5 times the diameter of the base for lean coals in sizes from ½ to ¾-inch lumps.

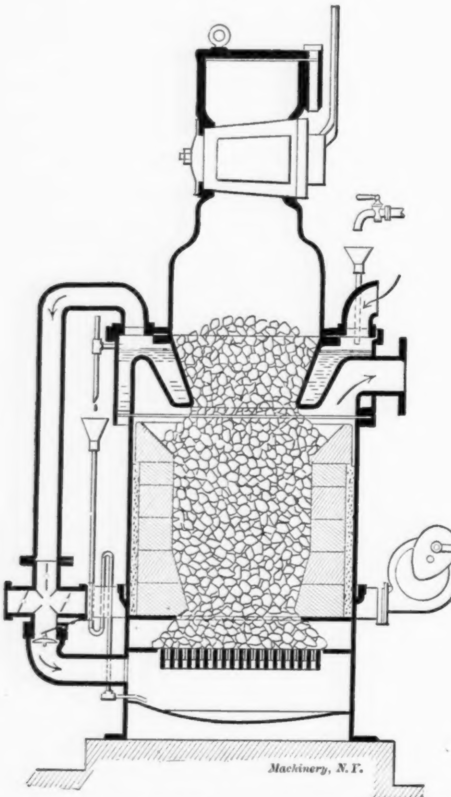


Fig. 1. A Sectional View of a Suction Gas Producer.

The composition of gas produced by various systems of suction gas producers averages as follows:

Carbon monoxide	24 per cent.
Carbon dioxide	5 per cent.
Hydrogen	17 per cent.
Nitrogen	54 per cent.

In selecting a fuel the lumps should range from ½ to 1¼ inches in diameter and the coal should be as free from dust as possible and contain not more than 8 or 10 per cent of ash in order that the grate may not be obstructed. The volatile matter must not be over 8 to 10 per cent and the coal must not coke or swell during the combustion, otherwise arches will be formed, impeding the natural descent of the fuel. Further to prevent this danger, the producer must be provided with poke-ing holes so that the fire may be stirred without admitting air. Coals containing sulphur or producing much tar should be avoided. If an excess of air enters the producer through the grate it interferes with the distillation of gas by cooling the zone of combustion and checking the formation of steam. If air enters above the fuel it dilutes the gas and may produce dangerous explosive mixtures. Hence all parts of the apparatus, including vaporizers, scrubbers, etc., should be tightly closed.

The preference is at present given to producers equipped with the internally heated vaporizer. Externally fired vaporizers, however, constructed with tubes having thin walls and a large amount of heating surface, heat the water more quickly and produce sufficient vapor to enrich the gas in from 10 to 13 minutes after starting the fire. With a well-designed tubular

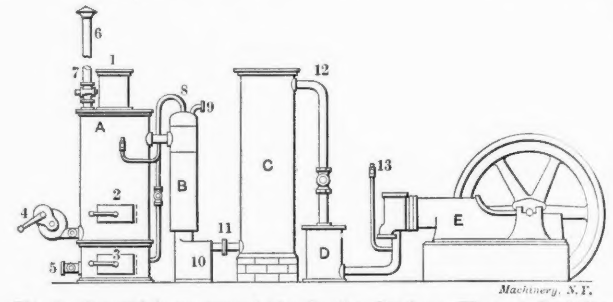


Fig. 2. General Arrangement of a Suction Producer Plant. 1, Coal Hopper with Gas Lock; 2, Cleaning Door; 3, Ash-pit Door; 4, Hand Blower, for starting; 5, Air Valve; 6, Escape Pipe; 7, Outlet Valve—these for use when the Gas Engine is not running. 8, Pipe conveying Steam from Vaporizer to Ash Pit; 9, Air Inlet to Vaporizer; 10, Water Box; 11, Gas Connection to Scrubber; 12, Gas Outlet from Scrubber; 13, Test Valve for Sampling Gas at Engine

vaporizer the producer and motor should be running at full power in 25 minutes after lighting up. Tubes of brass or copper should not be used in tubular vaporizers, as ammonia and sulphur corrode them. The scrubber, in accordance with good practice, is made six or eight times the volume of the producer, the height being three to four times its diameter. The best filling for the scrubber is metallurgical coke in lumps of three to four inches diameter. For washing the gas from three to five gallons pure water per horse power hour are required when the fuel is anthracite.

The above principles apply equally well to producers which supply motors of 10 horse power as to those of 150 to 200 horse power engines. Tests made upon a Körting suction producer and engine plant of only 6 horse power, gave a gross fuel consumption of 1.126 pounds of anthracite per effective horse-power hour. Complete tests made upon a double-acting Otto-Deutz engine and producer of 200 horse power gave the low consumption of 0.722 pounds of coal for an effective horse-power hour.

PHENOMENA OF MACHINE OPERATION.

John Richards, Journal of the Association of Engineering Societies, March, 1905.

The purpose of this paper is to call attention to the fact that the conditions of actual practice are often best met, not by machines figured out and determined from a drafting board from scientific data, but rather by such as have come to us through evolution and have been developed mainly by other means than computation.

In static structures, that do not involve machine motion, or that branch of constructive work we commonly call civil engineering, there is a close relation with science; means and agents are becoming uniform and can be computed, and results predicted, with much certainty. Strains can be defined; the properties of material are ascertainable; and extraneous forces, such as stress of the elements, the stability, oxidation and decay of material, and even its deterioration by fatigue, are becoming known and computable. In machine operation, however, the path is by no means so clear and perhaps never can be. Nevertheless, some of the general phenomena of operation are becoming susceptible of computation and scientific treatment; but, as I believe, to a much less extent than is generally assumed and believed.

The development of turbine water wheels may be taken as an example of the lack of real insight into the phenomena of machine operation. Such wheels were made the subject of research by eminent French engineers of the middle of the past century. These men, commissioned by their government, laid

down scientific rules to govern the construction of these wheels, and produced the three types of turbine water wheels known as the outward flow, the Jonval or parallel flow, and the impulse wheels of Girard. About 1850, two American engineers constructed at Lowell, Mass., what have remained, up to the present time, the finest example of outward flow turbines on this continent. The other types mentioned were later introduced into America. Here was a complete mathematical development of water turbines carried out to operate at the greatest efficiency. The subject of the water seemed ended, as in fact it had been so far as efficiency was concerned, but there was another phase to be dealt with in the operating conditions.

The French turbines were refined machines, expensive, and adapted for pure water. Gravel, driftwood, and other debris would not pass through the fine issues of the turbines, hence the latter were not well adapted to American conditions. Our American mechanics then began in an experimental way, whittling out new models. In the French wheels, the running expensive elements were outside, and occupied the extreme diameter, while the rough and inexpensive fixed elements were placed internally and were of relatively small diameter; the result in construction being expensive and requiring a slow rate of revolution with a strong and expensive transmission gear. Engineers, however, were so accustomed to associate centrifugal effect with turbines, that radial or outward flow seemed an essential condition; in fact, it had little to do with the case, as was found out by later experiment.

The American mechanics, after many years of "whittling" out models, succeeded in turning the wheels "inside out," or inverted them, so to speak, making the internal or smaller elements the running part, so that the water flowed inward toward the center, then changed its course 90 degrees downward in helical passages for escape. This was done entirely without scientific aid, in some cases even controverting scientific rules, and the result is the centripetal or inward flow turbine, the standard water wheel of this country, of which a single firm has made more than 10,000, and the wheels have even found their way back to France. Their efficiency is fully equal to, or even greater than, that of the older types, and the cost of the wheels is about one-half as great. This evolution has required about sixty years, and present practice rests mainly upon observed phenomena and upon the operating conditions rather than upon computed data. There was not even a draftsman in the works where the wheels were made that gained the highest award at the careful trials conducted at the Centennial Exposition, in 1876.

In the case of elastic fluids, impulse motors or steam turbines have been more than a century in evolution, notwithstanding that more than 400 patents have been granted in Great Britain alone for inventions pertaining to these machines, some of them a century ago and many of them fifty years ago. Mr. Parsons, who has been prominent in this work during later years, is, no doubt, one of the greatest living adepts in the science of thermodynamics, and, as is claimed, he has forecast with much accuracy the development of his turbine schemes as they progressed from 48 down to 11 pounds of steam for each horse power hour, but it is also claimed that he has expended half a million dollars in experiments. If inquiry were made, Mr. Parsons would probably admit that not one-fourth of his data came from computed sources, and that the observed phenomena of operation and adaptation have comprised the other three-fourths.

A wider and more important example of evolution in operating phenomena is furnished by piston steam engines. I do not mean the thermodynamic development of these, which is the greater part, furnished mainly by scientific deduction and experiment, but to the mechanical evolution of their operating parts, which had to keep pace with the thermal problems.

Down to twenty-five years ago it was a common object, in steam engine design, to reduce surface and velocity in bearings, partly to avoid friction, and partly because reduction of weight and space were also incentives, but the operating phenomena of machine bearings was a mystery in so far as any scientific rules were available.

Alignment, or the fit of bearing surfaces, especially in the case of cranks, is yet a mystery, if considered in a practical way. The most careful computations, respecting the flexure of shafts, frames, crank disks and pins, fail to disclose the operating phenomena. One has only to observe the center of an overhung crank or disk, even of the strongest proportions, to see that it describes a visible ellipse when under heavy strain and for reasons not explainable by computation. Similarly obscure operating conditions exist in various other parts of steam engines, and proportions are, beyond question, based more upon observed operating phenomena than upon computed dimensions.

Bearings that operate under steam, slide valves for example, were scraped to a perfect fit; cylinders were bored out with a smooth, glistening surface under a belief that such fitting was theoretically correct, but, by accident mainly, it was found that the bearing surfaces performed much better when they were not smooth and in perfect contact. A film of interposed water or oil produced the uniform fit.

In crushing hard material, such as quartz, with metallic surfaces, it was naturally inferred that the metal opposed to the stone should be as hard as possible, but, for reasons not easy to explain, soft metal endures longest. Cornish rollers are now covered with rings or tires of soft, fibrous iron. The sand blast discloses a like phenomenon. It is easier to bore a hole through a file with the sand jet than through a thin sheet of copper. An emery wheel will rapidly cut away tempered steel, but not soft iron. It is a problem of friability, no doubt, but is not fully explained. The whole field of mechanics is full of unexplained phenomena and mysteries, such as the temper of steel, the fatigue of metals, their crystallization under rhythmic concussion, the inherent strains in molded steel, the surge and reaction of moving liquids under high pressure.

Much that is written is apt to lead to the conclusion that scientific calculation alone suffices, in machine design, without the exercise of logical reasoning and practical observation of the operating phenomena and the conditions of use. Academic institutions should, at least, temper their theoretical instructions with the required warning that the phenomena of the operation of machines must be a principal factor in their successful evolution.

BELTING.

American Engineer and Railway Journal, May, 1905.

The following notes are abstracted from a set of belting instructions compiled by Mr. F. M. Whyte, general mechanical engineer of the New York Central Lines, and issued for use in the shops by Mr. R. T. Shea.

It is desirable to locate the machinery so that the belts shall run off from each shaft in opposite directions, as this arrangement will relieve the bearings from the increased friction that would result were the belts all to pull the same way. Two shafts connected by a belt should never be placed one directly over the other if possible to avoid it, as in such a case the belt must be kept very tight to do the work. It is desirable that the angle of the belt with the floor should not exceed 45 degrees. If possible the machinery should be so placed that the direction of the belt motion shall be from the top of the driving to the top of the driven pulley. The faces of pulleys should be about 25 per cent. wider than their belts. When practicable, belts should be tightened by moving one pulley away from the other.

The ability of a belt to transmit power depends upon the tension under which it is run, the degree of friction between the belt and the pulley, the complete contact of the belt with the pulley, the speed of the belt, and the arc of the pulley in contact with the belt. The tensile strength of single, ordinary tanned leather belting is about 4,000 pounds per square inch. The working strain should not exceed 10 per cent. of its tensile strength. The average leather belt will not transmit a force equal to its strength, for the reason that it will slip on its pulley before it will break.

As the friction of leather on leather is five times as great as that of leather on iron, the adhesion between the belt and the pulley can be greatly increased by covering the pulley with leather. The belt is thus capable of doing more work for a given width; the belt tension can be lessened to get the necessary friction, thus adding to the life of the belt; and unnecessary wear of the belt and a wasteful loss of power due to its

slipping on the pulley are prevented. The strain to be allowed for all widths of belting—single, light double and heavy double—is in direct proportion to the thickness of the belt, firmness of the leather being the same in all cases. Avoid running belts too tight, as great tension shortens the life of the belt, occasions a waste of power and causes great inconvenience from hot boxes, broken pulleys, and "sprung" shafting. Belts, like gears, have a pitch line, or a circumference of uniform motion. This circumference is within the thickness of the belt, and must be considered, if pulleys vary greatly in diameter and a required speed be necessary.

Belts are more satisfactory made narrow and thick, rather than wide and thin. Thin belts should not be run at a high speed nor wide belts be made thin. Such almost invariably run in waves on the slack side, or travel from side to side of the pulley, especially if the load changes suddenly. This waving and snapping wears the belts very fast; it is greatly obviated by the use of a suitable thickness in the belts. For new belts those that have already been filled with some good waterproof dressing are preferable to "dry" belts, for if not so filled they soon will be, with lubricating oil and water, a combination that will ruin any belt. Rubber belts should be used in places exposed to the weather, as they do not absorb moisture, nor so readily stretch or decay as leather belts under like circumstances. A new belt should be made straight, and if so made will run absolutely straight if the pulleys are in line. Slots punched in the center of a belt allow a chance for the air to escape between the belt and the pulley, and prevent "air cushion"; this is of particular advantage in all belts running at high speed.

It is safe and advisable to use a double belt on a pulley 12 inches in diameter, or larger. Light double belting runs steadily, with a minimum of "snap" or vibration, and does not twist out of place like single belting. It is successfully used for counter belts where shifters are used and where the work is not sufficiently hard to demand a heavy double belt; it is especially adapted for use on cone or flange pulleys, as it will keep its place and is less liable to turn over, and at the same time is pliable enough to hug the pulleys like a single belt. Double belting, light or heavy, is not recommended for twist belts at high speed, nor for wood work where belts are exposed to a large amount of chips or shavings, nor for places where much oil or water are liable to get on it.

As a means of making necessary alterations in the length of a belt the laced joint is recommended. To lace a belt, cut the ends perfectly true with the aid of a try-square. Punch the holes exactly opposite each other in the two ends. The grain (hair) side of belt should be run next to the pulley, and the belt should run off, not on to the laps. For belts 1 inch to 2¼ inches wide use ¼-inch lacing; 2½ inches to 4½ inches wide, use 5/16-inch lacing; 5 inches to 12 inches wide, use ⅜-inch lacing. For wider belts use wider lacing. Avoid thick lacing. In punching a belt for the lacing, it is desirable to use an oval punch, the longer diameter of the punch being parallel with the belt, so as to cut off as little of the leather as possible. There should be in each end of the belt two rows of holes staggered. Holes should be as small as possible. Recommended number of holes in the belt end for various widths are as follows:

Width in inches....	2	2½	3	4	5	6	8	10	12
Number of holes...	3	4		5	7	9	11	15	19	23

The edge of any hole should not come nearer to side of the belt than ⅝ inch, nor nearer the end than ⅞ inch. The second row should be at least 1¾ inches from the end of the belt. On wide belts these distances should be even a little greater. Begin to lace in the center of belt, and take much care to keep the ends exactly in line, and to lace both the sides with equal tightness. The lacing should not be crossed on the side of the belt that runs next to the pulley.

Belts and pulleys should be kept clean and free from accumulations of dust and grease, and particularly lubricating oils, some of which permanently injure the leather. They should be well protected against water, and even moisture, unless especially waterproofed. Resin should not be used to prevent belts from slipping. If a belt slips see first that the pulley is not dirty. Clean all the dirt from it and from the belt; rub the pulley surface of the belt with a dressing composed of 2 parts of tallow and 1 part of fish oil, rendered and allowed to cool before using. This will soften a belt and also preserve it, and it will not build up on the pulley and cause the belt to run to one side. If the belt then slips it is overloaded, and the remedy lies in a leather-covered pulley, a wider belt or a larger pulley.

A METHOD OF PREVENTING VIBRATION IN STEAMSHIPS.

From Paper by A. Mallock, Institution of Naval Architects, May, 1905.

In twin-screw ships, no matter how much the engines individually are out of balance, freedom from vibration in the ship can be secured if the engines are constrained to run at the same speed and in opposite phase to one another. I have often, in reports on such subjects, referred to the complete absence of vibration in twin-screw vessels, when the phases of the engine are opposed, as an illustration of the fact that properly applied balance weights would prevent vibration altogether; for in this case the moving parts of each engine act as balance weights to the other, and if any method could be found of keeping the relative phase of the two engines constant no other balancing would be required. Of course, it is impracticable to connect the engines by gearing or any equivalent mechanical device. Any connection between the two engines, if the method is to be a practical success, must be applicable at will and capable of being removed with ease without stopping the engines or interfering in any way with their separate working when removed.

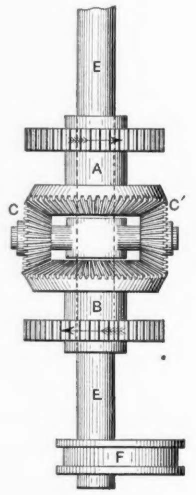


FIG. 1

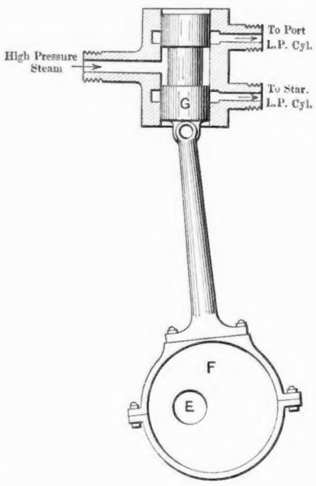


FIG. 2

Machinery, N.Y.

A Marine Engine Governor to Prevent Vibration in Steamships.

All this might be done with a governor of the form suggested diagrammatically in Figs. 1 and 2. A and B and C C_1 are a set of differential wheels, of which A is driven by the port and B by the starboard engine. The axes of the intermediate wheels C C_1 are fixed to the shaft E, which carries the eccentric F from which the valve G is worked. As long as A and B revolve at the same speed the shaft E remains stationary, but it rotates if the speeds of A and B differ. The valve at G is made so that if the shaft E, with its eccentric F, moves in either direction from some assigned position, owing to a difference of speed between A and B, high-pressure steam is admitted to the low-pressure cylinder of the engine which has the lesser speed until the valve and eccentric resume their original position. The steam pipes required for this purpose would be very small. By cutting off the steam supply to G the governor would cease to operate, and the engines would then be absolutely independent.

I have had a recent experience, on one of the largest of the Cape liners, of how annoying an intermittent vibration can be. In the ship in question the vibration was horizontal, and the engines were generally running at speeds differing by something less than a revolution per minute. Thus during intervals of rather more than a minute the vibration would cease and again rise to a maximum, and while being kept awake from this cause I had plenty of time to think if no remedy could be found less drastic than altering the engines them-

selves. I believe that the plan now sketched would be effective, and the cost, even for the largest ship, would be reckoned in hundreds rather than in thousands of pounds.

GAS PRODUCER POWER PLANTS.

Paper by Samuel S Wyer, Meeting American Institute Mining Engineers, at Washington, D. C., May, 1905.

The fact that gas producer power plants have received so little attention in America may be attributed to five conditions: (1) Ignorance and prejudice, (2) newness of work, (3) inadaptability of gas engines, (4) fuel economy not imperative, (5) smoke nuisance not given attention.

1. The only literature pertaining to gas producer power plants is that found in the various technical journals and in the transactions of engineering and other technical societies. In many cases the papers are of a fragmentary character, and seldom are they complete or comprehensive. It may be that the lack of reliable data available to engineers is the cause of the ignorance and prejudice that exists concerning this important branch of engineering.

2. The manufacture of producer gas is an old process, and gas engines have been developed to a very high stage of mechanical efficiency, hence there is no valid reason why such installations should be regarded as experimental.

The Winchester Repeating Arms Company, at its plant in New Haven, Conn., has a Loomis-Pettibone gas producer plant, built primarily to furnish gas for fuel purposes (such as for annealing ovens, furnaces, etc.); a 100-horsepower Westinghouse gas engine was installed some time ago, and later three direct-connected units, each of 175 horsepower, have been ordered. At the present time this example is one of the best instances in America of an industrial producer gas plant where gas is furnished both for fuel and for power.

The following list comprises some of the larger gas producer power plants now in operation in America:

Moctezuma Copper Co., Nacozari, Sonora, Mexico.

Guggenheim Exploration Co., 700 horsepower, Santa Barbara, Chihuahua, Mexico.

Detroit Copper Mining Co., 1,000 horsepower, Morenci, Ariz.

Rockland Electric Co., 1,000 horsepower, Hillburn N. Y.

Potosina Electric Co., 600 horsepower, San Luis Potosi, Mexico.

Velardeña Mining & Smelting Co., 2,000 horsepower, Velardeña, Durango, Mexico.

Sayles Bleacheries, 250 horsepower, Saylersville, R. I.

It is obvious that much has already been accomplished in this important field of power generation.

3. No gas producer power plant can be successful unless the gas engine is adapted to suit the particular gas available for its use. On the authority of Westinghouse, Church, Kerr & Co., an engine which will develop 100 horsepower with natural gas will give only about 80 horsepower with producer gas —a loss of 20 per cent. With a 200-horsepower engine this loss would be about 15 per cent, and with sizes above 300 horsepower it would be about 10 per cent. Hence, the obvious necessity of designing the engine to suit for the particular fuel it is to use. Several failures have been made by neglecting this important point.

4. In the list of plants given above, it will be noticed that most of them are in remote regions where the cost of fuel is high, hence the high economy of the gas producer plant was necessarily a feature that commended itself.

5. The laxity of the laws regarding the smoke nuisance has not made it imperative for manufacturers to give attention to the prevention of smoke. As soon as regulations concerning the smoke nuisance are enforced the gas producer industry will receive a new impetus on account of the easy solution that the gas producer plant offers for this trouble.

Data relative to the design, operation and maintenance of gas producer plants, mostly highly favorable to the use of such plants, are given as follows:

1. A good gas producer, from the very nature of its construction and operation, does not allow the smoke to escape into the atmosphere; hence the gas producer itself presents a practical solution for the elimination of the smoke nuisance. The non-requirement of a chimney means a large saving in the first cost and in the maintenance of a power plant, and is an additional advantage in plants where the æsthetic features of the design are of importance; for instance, in the case of a municipal power plant.

2. The cost of labor required to operate a gas producer plant is about the same as that required in a steam plant of similar size. However, during the time that a gas producer plant is idle it requires less attention than does a steam boiler. In the case of a municipal pumping station, the labor required to operate the producer gas plant would be one-half that of a similar steam plant, the gas plant being operated as follows: The gas producers to use coal for supplying the gas to operate a three-cylinder vertical gas engine direct connected to a triplex double-acting power pump. In this case the usual fire engine will be dispensed with, and, should a fire occur, the requisite pressure obtained by pumping directly into the system. For ordinary domestic supply the pump will deliver the water into a water tower, from which the mains receive the supply as needed. In every case the maximum quantity of water required during a fire is much larger than the average domestic consumption; hence the pump must be designed for this maximum quantity. As a result the working of the pump at its full capacity for six out of twenty-four hours would furnish enough water for the daily domestic consumption; the pump would usually be operated from 7 to 10 A. M. and from 3 to 6 P. M.

A gas holder of sufficient capacity to run the pump for thirty minutes is to be filled before the producers are closed down. Compressed air is to be used to start the engine, which may be put into motion simply by moving a lever. The engineer is to live adjacent to the plant so that when an alarm is sent in to the hose company and simultaneously to the engineer's home and to the plant it would be possible for the engineer to have the pump at work direct into the system by the time the fire company could reach the fire and make hose connections.

Since the gas holder would supply the engine until the producers could be started, the above scheme of operation eliminates the necessity of a night fireman and the keeping up of at least 70 pounds of steam pressure in a steam plant. A similar arrangement could be equally well adapted for fire purposes in connection with large industrial plants. With regard to the skill required, a producer gas power plant does not require any greater skilled labor than does a steam plant of similar size; however, in some cases it may require time for men trained to handle steam apparatus to become accustomed to gas engines and gas producers.

3. Two well-known engineering concerns give the following data regarding cost of installation:

The cost of gas power plants, including gas generating plant and gas engines, up to 500 horsepower, is about 25 per cent higher than the cost of a steam plant of similar size. Large plants, from 1,000 horsepower upward, cost about the same as a first-class steam plant of similar size.

4. The cost of repairs on a gas producer plant will not exceed that of a boiler plant.

5. In order that a gas producer plant shall be commercially successful, it must be able to make, from a low-priced fuel, gas that is sufficiently clean for use in an engine. Bituminous slack is usually the lowest priced fuel to be had; however, anthracite culm, or even wood, may be cheaper in some localities. In all cases the percentage of sulphur must be low if the gas is to be used in a gas engine. Frequently the use of a mechanically washed coal will be economical.

6. The only reliable way to remove tar and other hydrocarbons from gas made from soft coal is to have the producer so arranged that the gas comes in close contact with an incandescent mass of carbon. No mechanical means has yet been found to be successful, although several forms of centrifugal apparatus have been tried. For the removal of fine dust particles, however, centrifugal fans have proved very satisfactory.

7. The stand-by loss of heat is very small, being limited to radiation only; a gas producer is tightly closed during the time it is not making gas and the entrance of air is thereby prevented. This feature is a marked advantage over a steam boiler under similar conditions.

8. Even after a producer has been idle for several hours

it may be started and can be working at its full capacity within fifteen minutes. A gas holder is generally used in connection with the producer, from which a supply of gas can be taken to start the gas engine instantly and keep it in operation until the gas producers are making gas.

9. A gas producer may be stopped instantly by simply shutting off the supply of air and steam.

10. The gas from the gas producer is quite uniform in composition, and as it usually passes first to a holder before reaching the gas engine, it becomes thoroughly diffused, thus insuring a still greater uniformity.

11. The thermal efficiency of gas producers is generally about 80 per cent and in some cases it is even higher than this value.

12. It is much easier to use an automatic feeding device on a gas producer than on a steam boiler, because all producers are placed vertically and the fuel can be dropped into position by gravity. The use of an automatic feed always decreases labor and insures more uniformity in the composition of the gas produced.

13. The rate of gasification in a gas producer is relative to the character of the coal used. The best rate determined by experience is 12 pounds of coal per square foot of grate area per hour, although some makers have advised as high as 20 pounds of coal. Experience has also demonstrated that too rapid driving opens a wide door for the admission of adverse gasifying conditions.

14. The amount and frequency of poking a gas producer will depend on the nature of the fuel and the design of the producer. The mechanical agitation of the fuel bed (as in the Kitson and Fraser and Talbot producers) eliminates poking entirely. In using bituminous coals the difficulties of clinker formations is augmented by the production of coke. The judicious use of a steam blast and automatic feeding will generally reduce poking to a minimum and, in some cases, will eliminate it entirely. Hand poking is very laborious for the attendant and usually it will be shirked whenever possible. Gas will usually escape around the poke holes while the producer is being poked, which will vitiate the air in the producer room and also affect the regularity of the composition of the gas.

15. The calorific value of producer gas varies from 125 to 150 B. T. U. per cubic foot.

16. The generation of 1 brake horsepower per hour with from 1 to 1.25 pounds of coal or 3 pounds of wood is very common producer gas power plant practice at the present time, and the gas contains at least 80 per cent of the heat energy resident in the fuel.

17. A very important advantage of the producer gas installation is that the gas does not condense or lose power on its way to the gas engine. On the contrary, the cooler the gas the better it is for the engine. With steam the condensation is considerable.

18. It is easy to prevent leakage of gas from the piping, owing to the low pressure of the gas (about 2 inches of water); whereas, with steam, there is often much loss and inconvenience on this account.

19. By using isolated engines a large saving in shafting may be made in many cases. It is not possible to do this in steam plants and still maintain a good economy.

20. The floor space required for gas holders, gas producers, and auxiliary apparatus is about the same as that required in a steam plant; the holder, however, need not be placed adjacent to the producers, but at any other convenient place.

21. A gas producer plant is under much better control than the average steam plant, because in the gas producers the air supply rate of gasification as well as the fuel supply can be regulated more easily.

22. One of the most potent advantages of the gas producer plant compared with the steam plant is the ability of the former to store the heat energy in a holder where it may be drawn upon for immediate use. In this way irregularities and fluctuations of load need not affect the regularity of the action of the gas producer. This condition means an economy of operation and convenience of use that are impossible with any steam plant.

23. Another important advantage of the gas producer power plant is that, in many cases, the gas may be used both for power and for metallurgical purposes, the same pipes being used to supply engines and furnaces. The plant of the Winchester Repeating Arms Co., at New Haven, Conn., illustrates an installation of this character.

24. In many cases it is a serious matter to secure a sufficient supply of water for a steam plant and sometimes, even with an adequate supply, the quality of the water is such that it is entirely unfit for use in a steam boiler. One of the most annoying difficulties of many steam plants is the trouble caused by the corrosion and subsequent cleansing of the boilers, together with the maintenance of feed water purifiers.

The gas producer power plant forms an almost ideal solution for the problem of water supply. With a producer in normal condition, the consumption of water will not exceed 2 pounds per brake horsepower hour. The water used in cooling the gases in the scrubber may be cooled in a simple tower and used repeatedly.

25. There is no difficulty in piping gas for several thousand feet in order to reach an engine that drives an isolated machine; this often makes it possible to dispense with abnormal lengths of line shafting and the consequent friction loss or other unsatisfactory methods of power transmission. This condition is especially valuable in places where electrical power is not used.

26. Standard gas producers now range from a few horsepower to more than 500 horsepower in size.

27. There is less danger of explosion in a gas producer plant than there is in connection with a steam plant; moreover, should an explosion occur it would be much less violent and destructive than that of a steam boiler.

28. If desired, the gas producer plant may be placed near the fuel supply, which in many cases would reduce the expense of transportation, the gas being piped to the gas engines of furnaces where it is to be used. This arrangement, which is impossible with a steam plant, means a decided saving in favor of the gas producer installation.

29. The preceding paragraphs show the many strong advantages of the gas producer as a power generator; the large number now in successful operation shows that the experimental stage has been passed and that they have become a formidable competitor of the steam boiler. The time is not far distant when gas producer locomotives for railroad service, gas producer portable engines and gas producer power plants for marine service will be in common use.

The advantages of the gas producer for each of the above three classes are:

I. Gas Producer Locomotives, being—

1. *Smokeless.*—*a*, Trains and stations may be kept cleaner; *b*, tunnels may be passed through with greater safety; *c*, comfort of passengers will be increased.

2. *Cinderless.*—*a*, Fuel loss will be decreased; *b*, comfort of passengers will be increased; *c*, large fire losses due to sparks will be eminated entirely; *d*, insurance rates on property adjacent to railroads will be less.

3. *More Economical.*—*a*, In fuel, since the amount used would be less than one-half that used on steam locomotives; *b*, in water, since the amount used would be less than one-eighth that used on steam locomotives; *c*, in time, since the time required to take fuel and water would be less; *d*, in labor in firing on account of automatic feed and decreased amount of fuel used; *e*, in idleness, since stand-by losses would be very low; *f*, in number of fuel and water stations required.

4. *Safer*, since the danger of boiler explosions is eliminated.

II. Gas Producer Portable Engines, being—

1. *Smokeless.*—*a*, Large fire losses due to sparks will be eliminated entirely; *b*, insurance rates on property adjacent to where an engine is used would be less.

2. *More Economical* in, *a*, water; *b*, fuel; *c*, labor; *d*, time required to secure fuel and water.

3. *Safer*, the danger of explosion being eliminated.

III. Gas Producer Power Plants for Marine Service, being—

1. *Smokeless.*—*a*, Ships may be kept cleaner; *b*, passengers will have more comfort; *c*, a battleship could conceal its location more easily.

2. *More Economical* in, *a*, fuel; *b*, water; *c*, time required to fuel; *d*, bunker capacity; *e*, floor space; *f*, apparatus required, since all of the condensing machinery would be dispensed with.

CRANE MOTOR TESTING APPARATUS.

One of the interesting features of the big new Pawling & Harnischfeger plant on National Ave., Milwaukee, Wis., is the motor-testing apparatus in the crane motor department. Practically all the crane motors used in their crane installations are made in a well-equipped department for the work, at the east end of the shop. The motor-testing plant is at one end of the department, where a special switchboard and necessary apparatus have been installed for the testing work. Two testing machines are provided of different capacities in order to conveniently handle both large and small motors. Crane motors, being of the series-wound type, are tested for variable loads at variable speeds, the tests being conducted from no load to 50 per cent overload. The apparatus essentially consists of a generator mounted on trunnion bearings upon a cast-iron platen so that the generator frame can swing through a limited arc on an axis corresponding with the shaft axis. The motor to be tested is mounted on the platen in line with the generator shaft and is coupled to it by means of a flexible coupling, Fig. 3, which takes care of slight variations of alignment. To assist in aligning the motor with the generator shaft, a tool similar to a surface gage is used, having a point at a fixed distance from the platen which enters the center in the motor shaft when in the correct position. A tongue in the bottom of its base fits in the groove on the light projecting part of the platen seen in front, Fig. 1, and thus

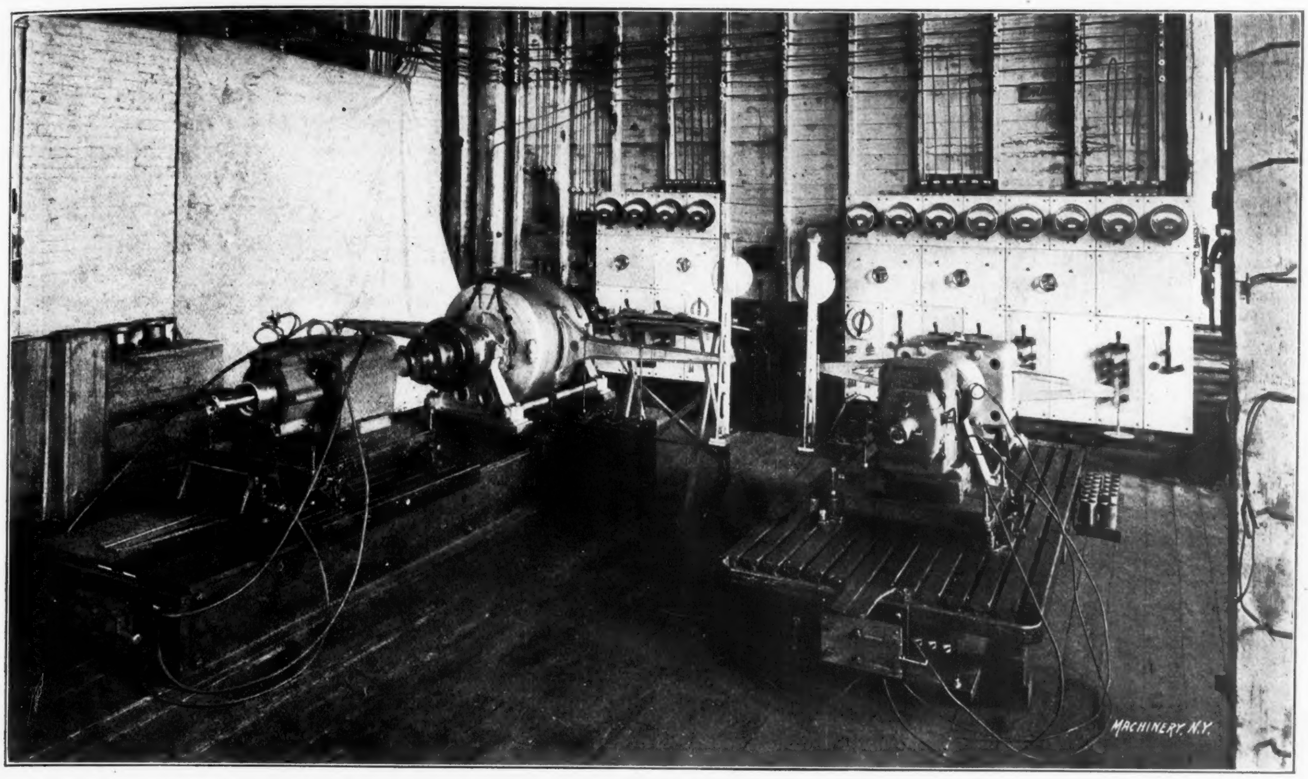


Fig. 1. View of Crane Motor Testing Apparatus, Pawling & Harnischfeger Shop, showing Switchboard.

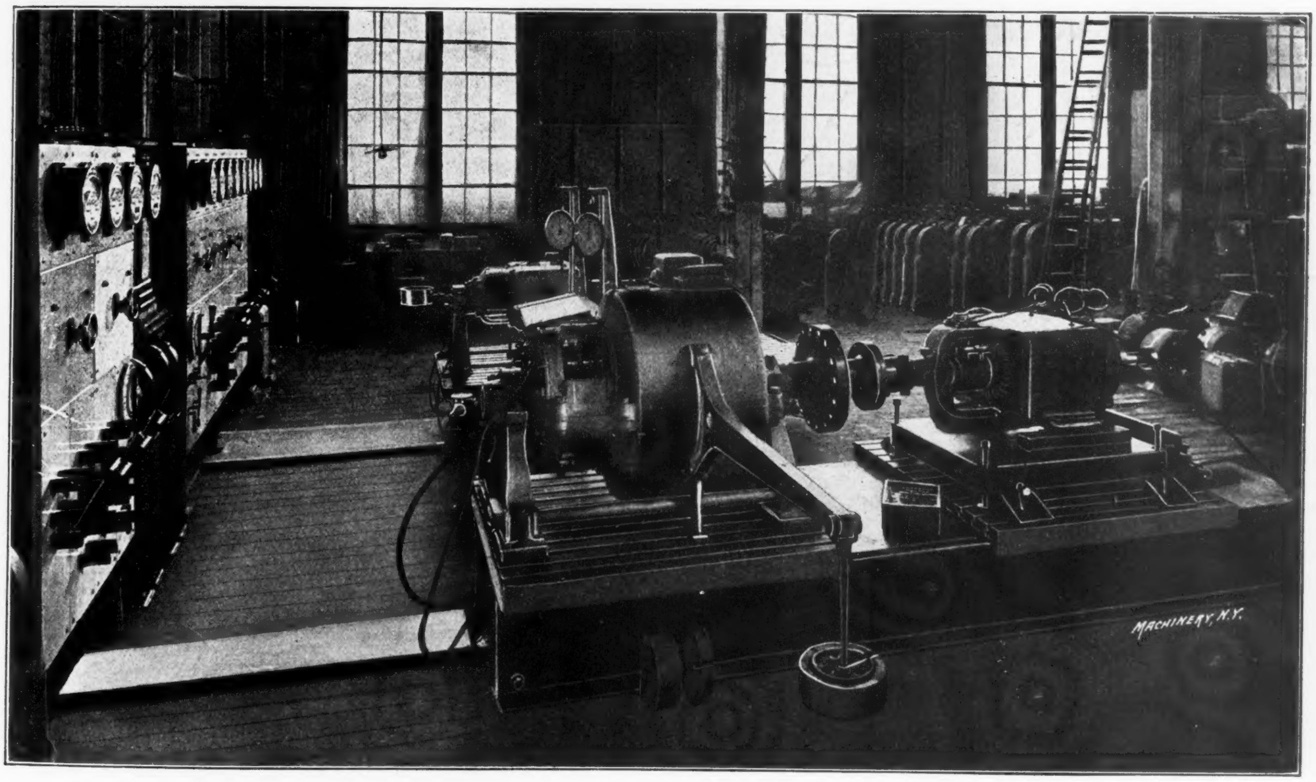


Fig. 2. Side View of Crane Motor Testing Apparatus, showing Counterbalance, Chatillon Scales and Tachometers.

provides a convenient and simple means of securing close alignment. The testing generator has two arms attached to the frame on opposite sides, to one of which is fastened a Chatillon weighing scale and to the other a counterbalance. The scale arm is made a specified length of 63⅛ inches from the center of the generator. With this length of scale arm a factor of 1/1,000 makes the H. P. formula for prony brake read $\frac{\text{R. P. M.} \times \text{pounds pull}}{1{,}000} = \text{brake H. P.}$ A Schaeffer & Budenberg tachometer or speed indicator is attached to the end of the generator armature shaft, thus giving direct readings of the speed at any moment. Hence, by observing the pull in pounds on the scale and the number of revolutions per minute, the brake horsepower may be instantly computed. For instance, if the revolutions per minute are 700 and the pull is 5 pounds, the brake horsepower is 3½ H. P.

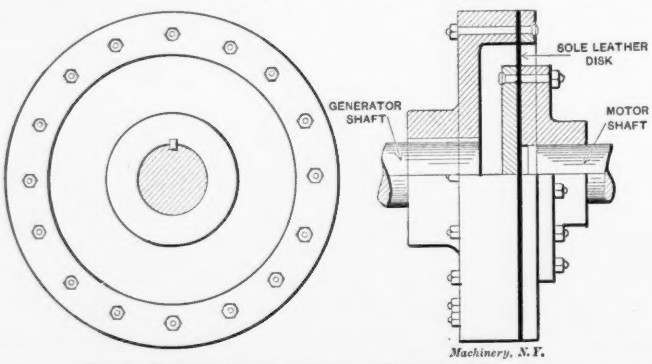


Fig. 3. Flexible Coupling between Motor and Generator.

The wiring for the testing services is of the parallel system. The motor to be tested is mounted on the platen and connected to the generator, and is operated at its normal speed. The field current of the testing generator is then regulated until the voltage in the latter is the same as the voltage supplied to the motor and then the two are switched together by starting box No. 2. The current for starting the motor and exciting the testing generator is supplied by the generator in the power house, the regulation of voltage being controlled from the testing department switch as shown by rheostat No. 3. The variation of motor load is obtained by regulation of the testing generator current through the rheostats Nos. 1 and 2, the latter also being arranged for series or parallel connection by small switches. From this it will be apparent to those familiar with electrical operation that the motor under test is driving the generator and, in effect, returning to the line the equivalent of the current consumed, minus the friction loss in the bearings, and the current loss in the two machines.

The apparatus is also adaptable for making reversed rotation tests of motors, it being essential, of course, that a crane motor should be reversible. The apparatus is also adapted to controller tests which are made by wiring the controller to a large capacity motor the armature of which is held by a specially designed brake, the brake being controlled during the test for any desired amperage. As each set of resistance is cut out on the controller, the volts and amperes are noted as shown by the instruments on the switchboard, and these readings are tabulated in a card index. In this connection it might be noted that all records of motor and crane tests are carefully kept in a very complete system of records. For the sketch, Fig. 4, illustrating the wiring of the apparatus, and the technical description of this apparatus, we are indebted to Mr. F. P. Breck, of the company's testing department.

* * *

TECHNICAL PUBLICITY ASSOCIATION.

An association has been formed of the various technical publicity departments of large manufacturing firms, especially those in the machinery field. Many manufacturers now have regularly organized departments devoted exclusively to disseminating information in regard to their products. Circular matter is prepared in these departments, data are supplied to the technical press, and advertising matter is made ready for the different journals. In these and other ways the publicity department keeps those interested informed upon the work of its firm, with results that are beneficial all along the line. Not only can more systematic work be accomplished, but these departments have able and technically trained men who prepare matter for distribution that rises above the grade of commercialism and becomes of real value to the technical reader, or to those in search of exact information in regard to different products.

There are now so many persons engaged in this important work that it has become possible to organize an association for their mutual good, and the first meeting was held in New York on April 27th, when an address was made by Mr. Emerson P. Harris, a well-known broker in publishing property upon "Machinery for Marketing Machinery."

Mr. Harris, whose familiarity with different publications is unusual, said that in no other department of publishing had the advertising medium risen to so high a degree of efficiency as in the technical field. The keynote of the ideal technical or trade paper is helpfulness to its readers. The greatest helpfulness depends upon a knowledge of the wants of a reader and sympathy with him on the part of the editor on the one hand, and on the readers by the confidence in the accuracy, reliability and truthfulness of the contents of the paper.

When these conditions between the editor and the readers are established, the technical journal undoubtedly becomes the

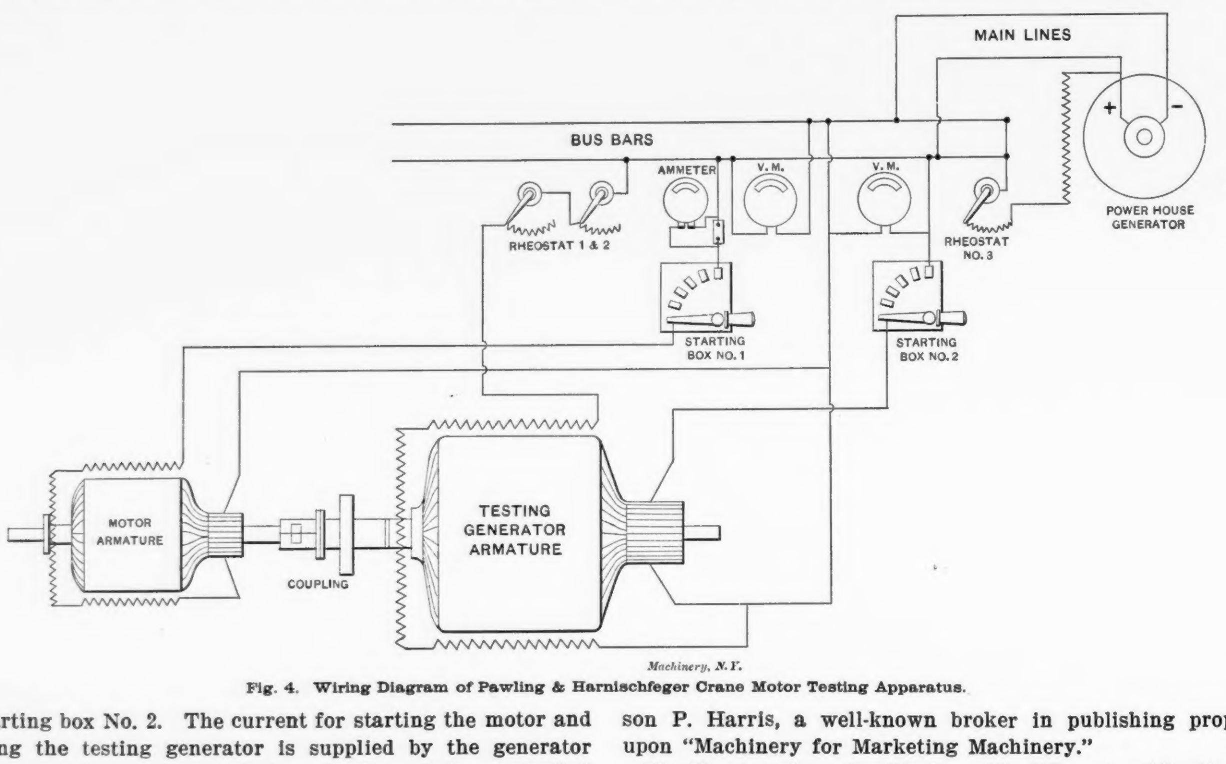


Fig. 4. Wiring Diagram of Pawling & Harnischfeger Crane Motor Testing Apparatus.

most effective advertising medium to the marketer of machinery. The mood which the editor inspires is carried over by the reader to the advertising page, and if it is one of confidence, and of interest in the new and important developments in the field of the publication, the advertisements are then looked upon with interest and in confidence. It is these psychological facts which make the best technical journals so efficient an advertising medium.

A question to be decided by the advertiser is whether a publication actually reaches the class of readers desired. No matters how good a paper is, it will not sell itself any more than machinery will. Circulation costs money. It requires much more in the case of technical papers than the circulation receipts begin to furnish the incentive for spending. It costs little to get a few subscribers in any field and to do business on these and a few sample copies—and "wind." Papers run on this plan can grant all kinds of accommodations in the way of puffs, for they have no faith to keep, nor character to lose. Such papers can make low rates but the space is likely to be had at any price.

The advertiser should know not only how many people pay for and read the paper, but he should have, as far as possible, an analysis of the circulation showing the classes of people who read the paper. This association would have an immense influence for good if it would not only insist upon having proof of circulation but, also, upon knowing how the circulation is obtained.

The speaker further commended the tendency toward concentrating advertising in fewer and stronger papers and using those papers intensely. Intensive is better than expensive advertising.

There were present at this meeting of the association the following members: Philip P. Kobbe, Rand Drill Co., President; H. M. Cleaver, Niles-Bement-Pond Co., Vice President; C. B. Morse, Ingersoll-Sergeant Drill Co., second Vice President; Geo. H. Gibson, International Steam Pump Co., Secretary; H. M. Davis, Sprague Electric Co., Treasurer; Rodman Gilder, Crocker-Wheeler Co., and Graham Smith, Westinghouse Companies, Executive Committee; H. T. Lauretzen, Holophane Glass Co.; F. S. Wayne, Robins Conveying Belt Co.; Lucius I. Wightman and Fred C. Iglehart, Ingersoll-Sergeant Drill Co.; M. C. McQuiston and J. O. Little, Westinghouse Companies; C. P. Hutchins, Jos. Dixon Crucible Co.; F. C. Cheston, American Wood Working Machinery Co.; Dean Park, Hammacher, Schlemmer & Co.; F. R. Matthews, De La Vergne Machine Co.; Dixon Boardman, Hall Signal Co.; H. F. Gale, General Electric Co.; C. S. Redfield and R. R. Glenn, Yale & Towne Mfg. Co.; A. E. Michel, International Steam Pump Co.; A. N. Barker, John A. Roebling's Sons Co.; W. B. Snow, B. F. Sturtevant Co.; E. F. Schaefer, Rand Drill Co.; H. H. Kress, Cameron Steam Pump Works; F. B. Vail, American Air Compressor Works; Arthur Warren, Allis-Chalmers Co.; A. L. Newton, I. T. E. Co.

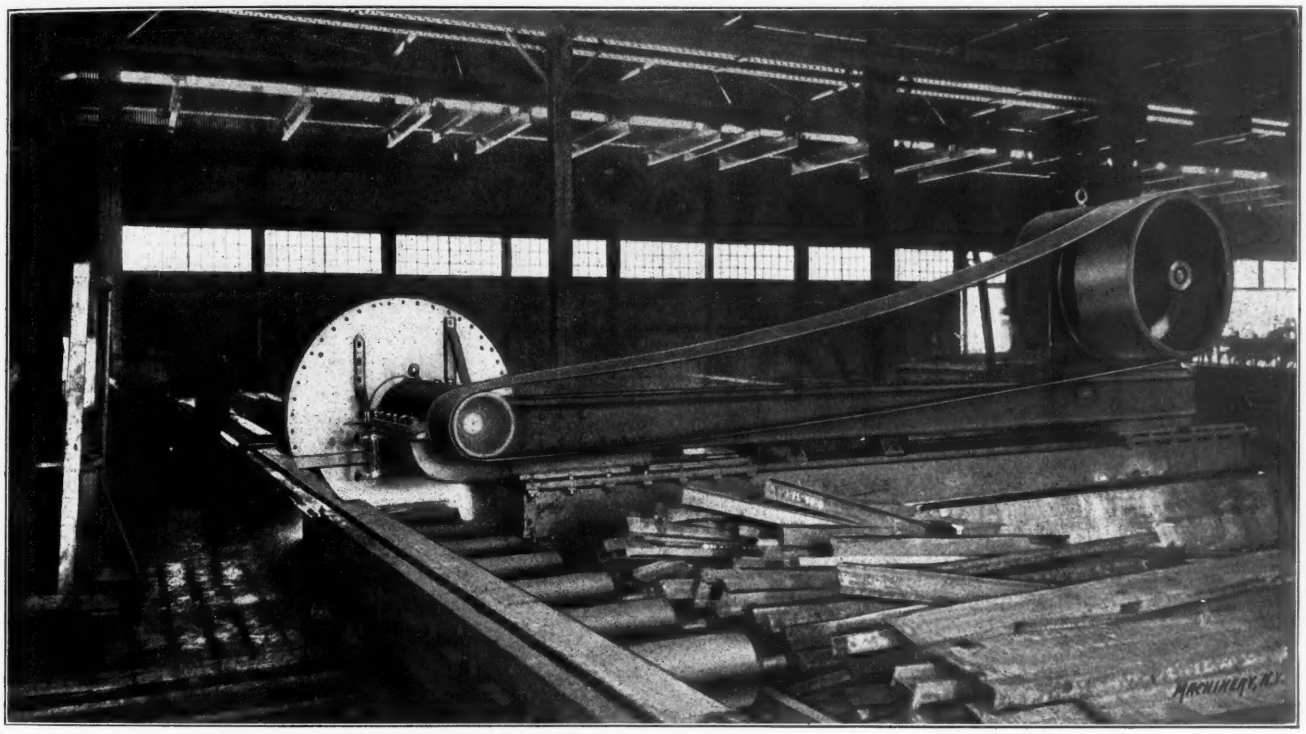
* * *

A LARGE FRICTION SAW.

The cutting of structural steel by friction saws has been done in the larger steel mills for several years, the rapidity of the cutting and adaptability for quick handling of material making this method superior to shearing or "cold sawing." The latest thing in this line of machines is a saw recently installed at the new 16th Street plant of Joseph T. Ryerson & Son, Chicago. This saw was designed by their own engineer, Mr. H. A. Ferguson, and was built by H. W. Caldwell & Sons Company, Chicago. It is of the type having a stationary bed of rollers, the saw moving across the work at a right angle.

The machine itself consists of four 12-inch 40-pound channels, 22 feet long, laid parallel and firmly secured by tie plates to the foundation. On top of these channels at each end are heavy castings with V-shape grooves and oilers. In these grooves slide the upper frame, also composed of four 12-inch channels, and having the shaft bearing on the front end, and the motor mounted on the rear. The shaft bearing is of the ring-oiling type, especially designed to take care of the high speed of the shaft and has an oil well capacity of fifteen gallons, dust proof. The motor is alternating current 100 horse power—220 volts at 580 R. P. M. It carries a drive pulley 48 inches diameter and a 16-inch three-ply belt. The pulley on the saw shaft is 14 inches diameter, giving a speed of 1,988 R. P. M. The saw disk is made of soft flange steel, ⅜-inch thick and 52 inches diameter, giving it a peripheral or cutting speed of something over 27,000 feet per minute. The upper part of the saw is covered with a shield of sheet steel, which prevents the abraded material being thrown above and back of the saw, and the operator, who stands in front, is protected from the flying sparks by a glass frame. An 8-inch hydraulic cylinder secured to the bed frame moves the upper frame back and forth across the work. The rollers for moving the material to be cut are 10 inches in diameter and power driven in two sections, one for bringing along material to be cut, and the other for taking it away.

The rapidity with which this machine cuts steel is simply marvelous. It goes through a 15-inch 42-pound I-beam in 9 seconds; a 20-inch 65-pound beam in 12 seconds; and a



Large Friction Saw in Joseph T. Ryerson & Son's Plant, Chicago, for Cutting off Structural Shapes.

24-inch 100-pound beam, or the largest size rolled, in 16 seconds. Its operation is accompanied with an ear-piercing shriek that is almost deafening, but the result is that the speeds of cutting off are far ahead of any previous records on cold steel.

* * *

ELECTRIC MOTORS IN STEEL MILLS.

The use of electric motors in steel mills was begun early in 1893, when two electric traveling cranes were installed, one in the Edgar Thomson Steel Works, Braddock, Pa., the other in Homestead Steel Works at Munhall, Pa., both of these plants being operated at the time by the Carnegie Steel Company. At that time, all machinery in mills was steam driven. The use of motors has proven so successful that to-day practically all auxiliary machinery is motor driven and mill engineers are now beginning to seriously consider the driving of the mills themselves by electric power. Only a short time ago an order was placed for two 1,500 H. P. 220-volt motors, which will be used for driving a rail mill at the Edgar Thomson Works of the Carnegie Steel Company. The use of motors, as large as these, may seem a radical step to many, but it is thought that, by introducing a flexible element between the prime mover and the load, which is of an excessively intermittent character, that there will be a sufficient saving in repairs to pay for the additional equipment within a few years. The maintenance on steam engines used for driving mills is, probably without exception, higher than in any other class of service.

In the application of electric power in steel mills, by far the most difficult problem presented, has been that of driving the main mill tables with motors. Many plants can be found where the "transfer," "run out," "shear" tables, etc., are all motor driven, but the main mill tables still steam engine driven. There are several reasons for this. To begin with, these rolls must be reversed at very frequent intervals, in many cases ten times per minute. The average operator cannot be depended upon to use any judgment whatever in the handling of the controller. This means that the motor is liable to receive enormous rushes of current, which are not only destructive to the motor itself, but the resultant jerks to the machinery in general are sufficient, in some cases, to twist off shafts, strip gears and do much other damage of a like character. Even if the operator were careful, when the motor is suddenly reversed, its counter electromotive force causes it to act as a booster in series with the generator and thus, at the moment of reversal, there is almost double the line potential across the starting resistance and the motor consequently receives almost double the first rush of current which it should receive. This may seem to be merely a question of the proper design of the starting resistance, but if we stop to consider what it means to double the resistance in the ordinary manually operated controller, it will at once be seen that such a scheme is unfeasible, due to the increased drop across the various steps of resistance and the consequent sparking on the contacts.

To sum up the situation, it will be see that, in order to secure satisfactory results from motors used on this class of service, that the flow of current to the motor must, at all times, be automatically limited to a pre-determined maximum value. It is at once apparent that the operator cannot be depended upon to secure this result by the proper handling of the controller, and so there is but one expedient remaining to consider, a controller which will accomplish this result automatically. In other words, a controller, which might be termed an intelligent self-starting device.

The Cutler-Hammer Manufacturing Company have just installed in the new rail mill at the plant of the Republic Iron & Steel Company, Youngstown, Ohio, a number of controllers, which seem to fulfill the requirements for this class of service. These controllers are installed in connection with motors driving tilting tables, catcher's tables, etc. They consist of a drum type reversing switch and a number of solenoid controlled switches for cutting out the starting resistance. The drum type reversing switch, which is used to handle the main current of the motor, is equipped with powerful blow-out magnets. The accelerating switches are mounted on a slate panel carried on a switchboard frame. The operation of these controllers is as follows:

The operator closes the circuit to the motor on the drum type reversing switch for either one direction of rotation or the other and the motor is thrown across the line in series with all of the starting resistance. At the same time, the circuit of the solenoid switch, controlling the first section of accelerating resistance is closed. This switch in closing cuts out a section of the starting resistance and at the same time closes the circuit of the switch controlling the next section of starting resistance and so on through the entire progression of switches. In conjunction with these accelerating switches is a series relay solenoid, which may be adjusted to pick up on any predetermined current. This relay is so connected with the accelerating switches that, when it lifts, no more accelerating switches can close. Those switches that are already closed, however, are maintained in that position. In other words, the process of acceleration is halted until the current has fallen below the predetermined value. By using solenoid switches for cutting out resistance, all sparking is eliminated and sufficient resistance may be provided to hold down the motor current even when it is suddenly reversed and its consequent booster action produced.

On a test made on a 75 H. P. motor driving a tilting table, the motor was brought from full speed forward to full speed reverse in three seconds without the motor current at any time, exceeding 125 per cent of full load current. With such a controller in use, the use of motors on mill tables appears perfectly practicable, and no doubt, the next few years will see the steam auxiliary engine driven from its last stronghold in the steel mill. Perhaps, the next ten years will see the use of the steam engine for direct driving entirely dispensed with. The results of the above mentioned installation at the Edgar-Thomson Steel Works will, no doubt, be the deciding factor in this regard.

In connection with the foregoing it is of interest to note that the Cutler-Hammer Company has just received an order covering two 1,500 H. P. 250-volt automatic release starters for use in the Edgar Thomson Steel Works. These starters are particularly interesting, not only on account of the size, but on account of the general arrangement of the starter, the unique type of resistance used, and the large overload capacity of the equipment.

The starter proper will be built in the form of a switchboard of white Italian marble, approximately 7 feet 8 inches high and 11 feet long. The starter parts will be of the well-known type made by the company where the several sections of starting resistance are controlled by a number of independent levers. For the starter in question, the levers which will carry the motor current continuously are to have a continuous capacity of 10,000 amperes, and are guaranteed to carry this amount with a temperature rise not exceeding 40 deg. C. The intermediate starting levers, which as occasion demands, may also be used temporarily for regulating duty, will have 6,000 amperes capacity on the same basis.

The switchboard will consist of five panels, each panel containing two levers. The connection between the different panels will be made at the rear of the switchboard by bus-bars in the usual manner. At the front of the switchboard, such arrangements will be made that with very little trouble an entire panel may be removed from the board without disturbing the other panels, or the parts mounted on them. It will even be possible to operate the starter with one of the intermediate panels removed. The levers on the several panels are mechanically interlocked in the manner adopted for the multiple switch type of starters, so that the several levers composing the starter cannot be operated except in a predetermined order. As giving some idea of the large size of the starter levers, the overall length of each of the levers is a trifle over 3 feet, but by means of a toggle joint action these heavy switches may be closed with a remarkably small amount of force.

The resistance for these starters will consist of standard steel rails, mounted in a steel rack, approximately 20 feet high, 14 feet wide and 27 feet long. There will be about one hundred 25-pound rails and two hundred 40-pound rails used. This enormous capacity of resistance material is provided so

that the resistance may be used to control the speed of the motor for short periods of time, even under comparatively heavy loads without permanent injury. The main connections to the starter will consist of six 1,000,000 circular mill cables, while the intermediate resistance leads will consist of four 1,000,000 circular mill cables.

The motors which these starters will be used to control are of the Westinghouse make, and will be very heavily compounded. The motor armature shaft will carry a flywheel 20 feet in diameter, weighing approximately 125,000 pounds, and will be directly connected to the standard blooming mill rolls. The motors will be designed to stand very heavy overloads for short periods of time, and mechanically constructed so as to stand the enormous strains encountered in this service. The normal speed of the motor will be 100 R.P.M., and allowances will be made so that this speed may be increased 25 per cent by weakening the shunt field of the motor. If the installation of these motors proves satisfactory, which result is confidently expected, the Carnegie Steel Co. will equip the mills in its other plants with motors instead of the cumbersome steam engine now used.

For some time, the Pittsburg Reduction Co. has had a 1,500 H. P. 500-volt motor driving a blooming mill in satisfactory operation at its plant at Messina, N. Y., which is controlled by a 500-volt starter, similar in construction to those to be installed at Braddock. Since installation, this particular equipment has given every satisfaction, and it would now seem as if the motor drive was considered perfectly feasible for the largest mills.

* * *

ERRATUM.

A correspondent, Mr. R. S. Wright, of Sherbrooke, Quebec, Canada, calls our attention to an error in the article on Steam Hoisting Machinery by Mr. A. M. Levin in the May number of MACHINERY. The formula on page 455 of that issue for the normal pressure exerted by each beam of a post brake should read $N = \frac{n\,l}{m\,a} W$ instead of $\frac{1}{2} \frac{n\,l}{m\,a} W$ as it appears in Mr. Levin's article. This is evident from the following consideration, referring to figure No. 3: the weight W produces a turning moment about the fulcrum equal to Wn, $\frac{1}{2}$ of which is taken by each of the vertical rods. Hence the pull in each rod is $\frac{1}{2}\, Wn \div m$. This in turn produces a turning moment about the fulcrum at the top and bottom respectively of the brake beam equal to $\frac{1}{2}\, W \frac{n\,l}{m\,a}$, bringing a pull on each of the horizontal rods equal to $\frac{1}{2} W \frac{n\,l}{m}$. Hence the total normal pressure of each beam against the face of the brake is the sum of the pulls of these two horizontal rods, or $N = \frac{n\,l}{m\,a} W$.

Formula fourteen thus should read $L\,R < 2\,f\,r \frac{n\,l}{m\,a} W$ and (15) becomes $W > \frac{L\,R\,a\,m}{2\,f\,r\,n\,l}$.

* * *

THE "CORE AND COIL" PROBLEM.

In an address, "Unsolved Problems in Electrical Engineering," delivered by Col. R. E. B. Compton at the April 10, 1905, meeting of the Institution of Civil Engineers, attention was called to the "core and coil" problem, and the defects of the present system of winding coils. "Winding is done in three ways: "First with round wire wound with the wires of one layer substantially over the wires of the layer beneath. The second is also a usual form, and allows of a more solidly wound coil—that is, when the ends of the winding are stepped back, the succeeding layers fitting in some extent, between the depressions of the previous layer. But the third case constitutes an ideal winding of a coil to produce the same strength of magnetic field, and to give sufficient cooling surface to get rid of the heat, but using less copper, and having greater heat conductivity on account of the continuity of the copper from the inside to the outside of the coil. This ideal winding consists of thin copper strip wound on its edge into correct position by suitable machinery, each turn being insulated from the adjacent turn by a thin film of insulating material applied to it at the time it is being wound. This ideal form of coil winding has already been occasionally used in practice for the larger coils required for certain transformers, and for the magnets of large generators and motors; and it will be seen that we gain greatly in reducing the bulk of the winding, weight and cost of copper; and as the magnet arms are shorter, we gain greatly in the size and cost of the frame work of the machine; but in order to apply it generally to the smaller class of apparatus we still require a combination of rolling, winding, and insulating machinery which will take the rough copper strip and at one operation reduce it to the correct cross section, wind it into place, and apply the coating of insulating material. The limit of temperature which insulation should continuously stand should be increased to at least 390 degrees F., and if we are successful in obtaining the two requirements which I have mentioned, viz., the reduction in the sizes of our coils, by means of the last form of winding I mentioned, and the utilization of the higher temperatures at which we can work them without endangering the durability of our plant, we shall benefit greatly, both constructors and users, by reducing the bulk, weight and cost of our machinery, without in any way impairing its efficiency."

* * *

JUNE MEETING OF THE A. S. M. E.

The semi-annual meeting of the American Society of Mechanical Engineers to be held at Scranton, Pa., takes place from June 6 to 9 inclusive. The subjects of the papers to be presented are as follows: "The Transfer of Heat at High Temperatures" by Frank C. Wagner; "Standard Unit of Refrigeration," by F. E. Matthews; "Formation of Anchor Ice and Precise Temperature Measurements," by Howard T. Barnes; "Some Types of Centrifugal Pumps," by Wm. O. Webber; "Microstructure and Frictional Characteristics in Bearing Metals," by Melvin Price; "Cast Iron, Crushing Loads and Microstructure," by Wm. J. Keep; "Smoke and Its Abatement," by Chas. A. Benjamin; "Can a Steam Turbine be Started in an Emergency Quicker than a Reciprocating Engine of the Same Power?" by A. S. Mann; "Notes on Efficiency of Steam Generating Apparatus," by A. Bement; "Performance of a Superheater," by A. Bement; "Counterweights for Large Engines," by D. S. Jacobs; "Steam Actuated Valve Gear," by Wm. H. Collier; "Notes on Heads of Machine Screws," by H. G. Reist; "Belt Creep," by W. W. Bird; "Function of Laboratory Courses in the Curriculum of Engineering Schools," by Chas. E. Lucke; "Continuous Measuring and Mixing Machines," by E. N. Trump; "Epochs in Marine Engineering," by Geo. W. Melville.

* * *

Something over two years ago it was discovered in the cement testing laboratory of the C. M. & St. P. Ry. that a Portland cement briquette two years old was disintegrated by drippings of signal oil. The importance of the discovery caused a series of tests to be made to find out what the effect of oil was on Portland cement generally, and what could be done to prevent deterioration of cement structures which were unavoidably exposed to oil drippings. A report of these tests is given in the *Engineering Record*, but the investigation is of a negative character so far as finding any preventive. Painting with linseed oil paint delayed the action somewhat, but it eventually succumbed. Paraffine and sodium silicate were used with no better results, as was also the case with Sylvester's process for making cement impervious to water, which consists of alternate washes of 5 per cent. solution of alum and 7 per cent. solution of castile soap. In view of the fact that concrete construction is being largely used for roundhouses, oilhouses and other structures exposed to oil drippings, it seems highly necessary to find some means for correcting this fault. It is altogether probable, however, that mere oil spattering would not seriously deteriorate a concrete floor for a long period as the oil would probably be more or less oxidized before penetrating to any great depth.

LETTERS UPON PRACTICAL SUBJECTS.

GAGE MAKING.

Editor MACHINERY:

Possibly there is no branch of the tool-making trade that demands the skill and accuracy that gage making does. For one reason or another we seldom see anything in print regarding this exacting line of work; one reason is that a gage maker might enter very carefully into detail as to the manner in which to make this or that gage, but said detailed description would not apply to the method employed in the next shop. The object of this article is to touch briefly upon different methods and gages.

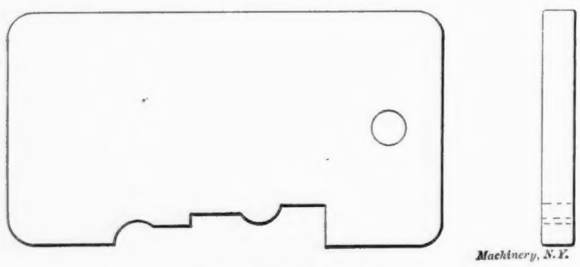


Fig. 1. Profile Gage.

It is becoming the general practice to make gages from machinery steel and caseharden same. As a small fraction of 0.001 inch renders a gage useless as a standard, it is therefore obvious that machinery steel hardened to a depth of 0.003 to 0.005 inch answers fully as well for gages as tool steel. But to obtain best results from plug and ring gages they should never be made of machinery steel. First, the gage may spring slightly during the hardening process, and as the gage is only hardened to a depth of 0.003 inch or thereabouts, the reader can readily see that the casehardening can easily be lapped away, leaving soft spots in the gage; this not only shortens the life of the gage but said soft spots "charge" with

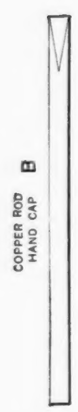


Fig. 2.

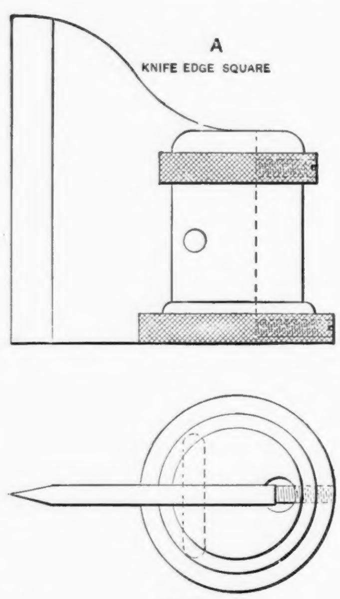


Fig. 3.

emery when lapping the gage to size, making the finished product half gage and half lap. But for snap gages and profile and receiving gages machinery steel is superior to tool steel, due to the fact that the gage is not distorted to any extent during the hardening process. The writer has found by experience that best results are obtained, when making plug gages, by using stock somewhat larger than finish gage size. For instance if the plug gage is to be 1 inch diameter, make same from a bar of steel say 1⅛ inch or larger. By so doing the scale and outer stock that has apparently been decarbonized to a certain degree is entirely removed. This same point is applicable to reamers, mandrels, dies and numerous other jobs that require hardening. To go further, let us suppose that a plug gage 1 inch diameter is turned up from a bar of steel slightly larger than 1 inch. After hardening it will be noticed that there are spots that seem to bulge; said spots are hard, but the surrounding stock is apparently soft. But, if the gage is ground to say 15-16 inch diameter, it will be found hard over its entire surface.

The methods of making gages vary greatly in different shops, *i. e.*, some manufacturers are content with gages turned and filed nearly to size, which after being hardened are polished to size with emery cloth. In the next shop we find the

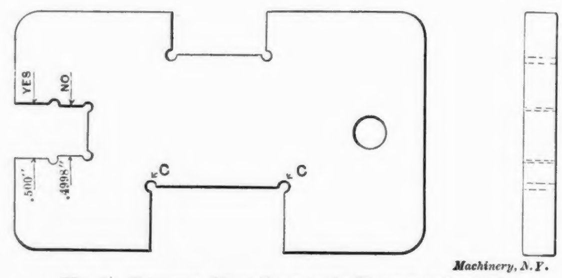


Fig. 4. Common Snap Gage with Clearance for Lap.

gages are hardened and ground to size. The next shop is more exacting and the gages are ground and lapped. Going still further we find manufacturers who are still more exacting, and demand that gages must be hardened, roughly ground, aged, finish ground, lapped, and the minute ridges caused by circular lapping, entirely removed by lapping the gage lengthwise to size. About 0.0001 inch is removed by this operation. The laps and manner in which steel is aged was described in the May, 1904, issue of MACHINERY.

When making a profile gage, Fig. 1, it is a good plan to first make a sheet steel templet to fit model perfectly. A planer tool is now fitted to the templet and the impression planed through three gages at the same setting. (The three gages referred to are master, inspector, and working gages.) Should the profile be of such a size as to render it impracticable to plane the entire surface at once, a series of formed tools are

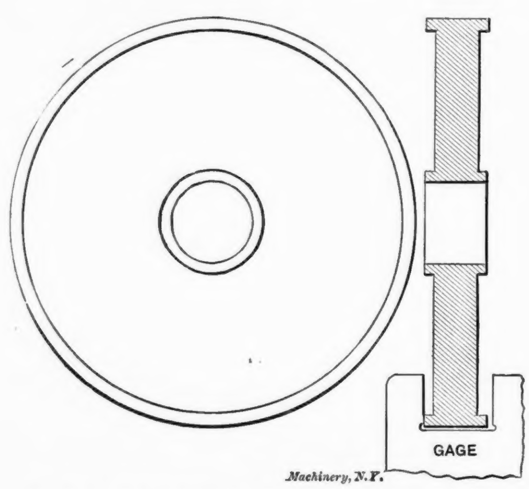


Fig. 5. Cast Iron Lap adapted to Lapping between Jaws.

then made, also a male templet, and the impression is planed to fit said templet reasonably close and then finished by hand. As it is absolutely necessary that the profile be the same over its entire surface, the knife-edge square, Fig. 3, will be found exceptionally well adapted for this work. The gages, after hardening, are lapped by hand to fit model by means of a flattened copper rod, Fig. 2, and flour emery. Should the gage "open a trifle" during the hardening process, the vise will prove an admirable "putting on tool," as the interior of gage is soft.

The common snap gage, Fig. 4, is carefully machined to within 0.002 inch of finish size, care being exercised that the

faces are smooth. The holes *C C* are made to allow clearance for lap, Fig. 5, which is a cast iron disk. The gage is nicely casehardened, and gripped in the vise of an old milling machine, preferably a hand machine reserved for this operation. The lap is placed on the arbor, smeared with emery paste and set in motion. By moving the table back and forth the gage can be lapped until the model can just be started to enter. The gage should then be finished by hand. If the gage is made to dimensions instead of a model, it is a good plan to make a temporary gage of drill rod, and fit the gage to same.

Fig. 6 shows a very simple style of snap gage, one that is easily duplicated. This gage requires the spacers *E*, which are made the required size, that is, the size of the piece to be

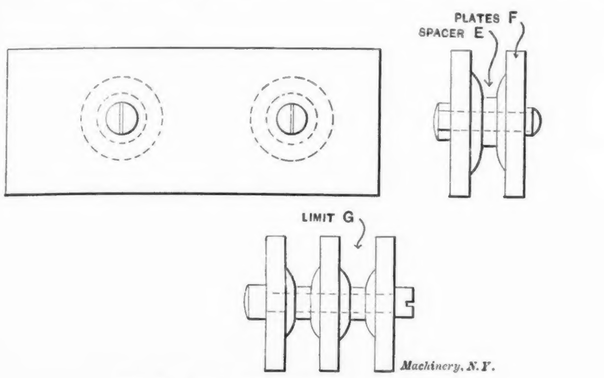


Fig. 6. Easily Duplicated Snap Gage.

gaged. The plates *F* are parallel pieces of hardened steel that have been ground and lapped. When the gage becomes worn, all that is necessary to duplicate the original size is to remove the plates and lap the surface true. A limit gage can be made by adding another plate and spacers of proper length, as at *G*.

The receiving gage, Fig. 7, is a very difficult gage to make and on account of the cost it is rarely used except where it is absolutely necessary to do so. The gage is made to fit perfectly the entire profile of the piece to be gaged, and is made of a series of small pieces fitted together, the object being to overcome as far as possible the distortion of steel when passing through the hardening process. The base *H* is of machinery steel casehardened, and its upper surface lapped perfectly level. The pieces *J* are ground and lapped on bottom, and the formed edges are lapped by hand to fit model. To obtain best results fit the pieces *J* to model while pieces are soft and fasten pieces to the base *H* by screws. The dowel holes are now drilled and tapped with a fine pitch tap, say 5-16 x 32. After pieces *J* and base are hardened, soft steel screws are

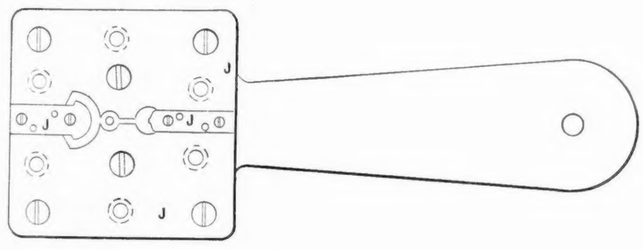


Fig. 7. Receiving Gage for Pallets.

turned securely into these holes and dressed off flush with top and bottom of pieces *J* and base. After the pieces are lapped to fit the model, they are tightened in their places, and the soft screw bushing drilled and reamed through for dowels. It is impracticable to attempt to lap dowel holes true, especially when they are tapered and do not line up. This soft screw bushing will be found useful on many other tools where dowel holes are apt to change during hardening.

Fig. 8 shows a universal snap gage that is designed especially for large work. All that is necessary to make one gage cover the field is to set the gage to required diameter (from standard length rods, so that pointer stands at zero) then as the gage hangs on the piece to be gaged, it is swung so that anvil *L* passes over highest point, and the pointer will record 200 times greater than what the error really is. Any one who has used a large micrometer for measuring such work as is found in arsenals knows the diagreeable manner in which one is obliged to obtain measurements. One man will

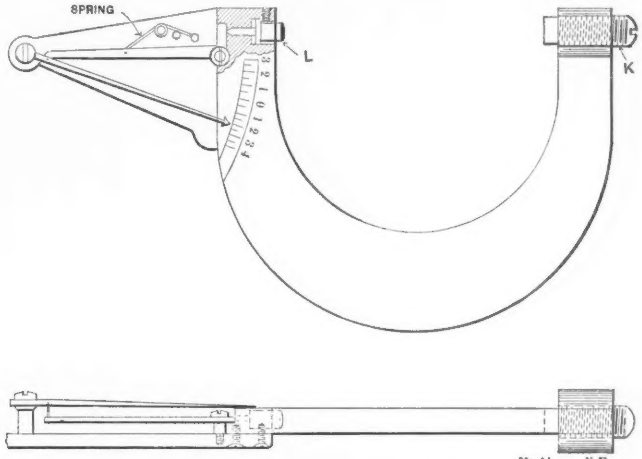


Fig. 8. Universal Snap Gage for Large Work.

hold the micrometer on the breech of a large gun and another man will do the measuring. With this form of gage one man can measure very handily, as all that is necessary to do is to note the number of graduations traversed by pointer when gage passes over work. F. E. Shailor.

Great Barrington, Mass.

AN ADVERTISING SCHEME.

Editor Machinery:

One of the problems of modern business is advertising—to bring goods to the notice of people likely to need them and to do it in such a way as to impress possible customers and cause them to remember that John Smith makes the Eureka hash grinder, and that it is the only hash grinder worth having, etc. To obtain this end a great variety of methods are in use, from obscene literature and vulgar devices, to works of art and things of great usefulness; in fact almost every one adds materially to his education by reading advertisements. It must be a source of pleasure to a good man to feel that his advertisements are not only increasing his business, but are an actual benefit to humanity and to such men I wish to suggest the following plan:

Wherever there is machinery there is always danger of some one getting injured, and in case of an accident few people understand how best to treat the sufferer, and again, very few factories have at hand any drugs, bandages, etc., so necessary in such cases. So quite often the unfortunate one has to suffer more than is necessary before medical assistance can be obtained, and not the least of his sufferings is the feeling of helplessness and uncertainty produced by the inability of those about him to help him.

Now my plan is for some one who has an article he wishes to advertise in manufacturing establishments, to furnish a small outfit of drugs, bandages, etc., selected for the purpose, and with each outfit several large placards, with the request that these cards be placed in prominent places about the factory where every one can read them. These cards could carry the advertisement of the one sending them, also simple instructions from a reliable authority explaining what to do in case of accidents of various kinds. Almost everyone has read such instructions (more or less reliable) at some time or other, but when the time comes to use them we find we have forgotten them; but with these cards constantly about the factory, some will become familiar with their contents or at least someone will have read them recently enough to know what to do when the accident happens. Now I hope some one will take up this plan and perhaps improve upon it, and I believe some one will, not only for his own good, but for the good of humanity. R. Earl Weinland.

Indianapolis, Ind.

SIX-INCH MICROMETER SURFACE GAGE.

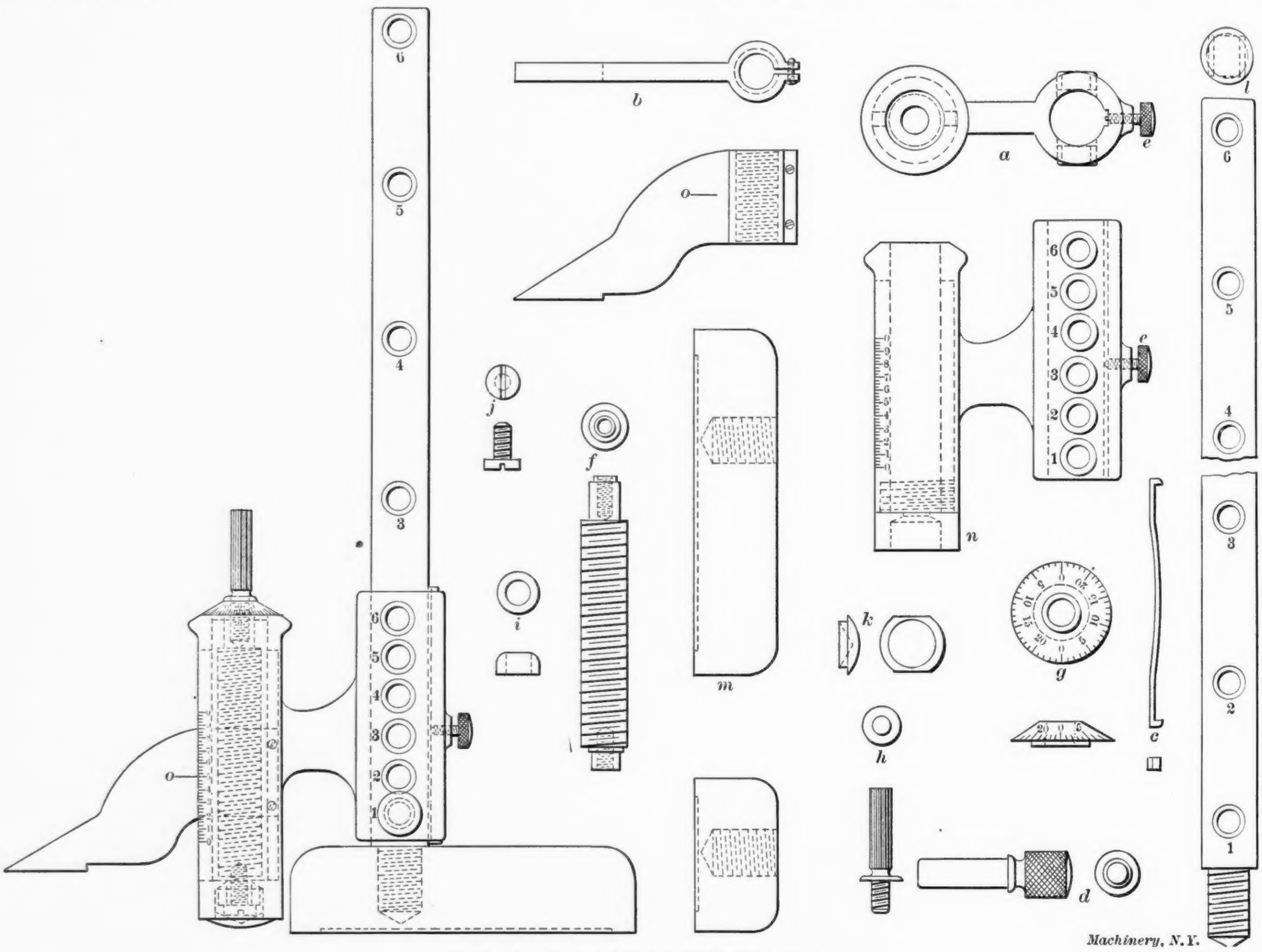
Editor MACHINERY:

MACHINERY is looked over every month by many who are seeking for new ideas in toolmakers' gages, etc., and I must confess to being guilty in ths respect, also. We toolmakers as a class are particularly interested when a tool described is not on the market, for we then know that we may be of a few to make such a tool to add to our collection. The tool shown herewith is one of my own design, and having used it successfully on various kinds of work, I can recommend it as an appropriate addition to any toolmaker's collection.

The tool to be described is a 6-inch micrometer surface gage, and is shown in the cut, which includes the assembled parts and the details. It can be quickly set to exact positions, varying by 1 inch from 1 to 6 inches in height, by inserting a hardened plug. A valuable feature of the tool is the set of six independent holes through both the movable part and the beam. Each hole is bushed with a hardened steel bushing, ground and lapped to fit the plug which locates to exactness the various inch settings.

inch wide to fit the spring gib, *c*, which holds the slide in position when the lapped plug, *d*, is withdrawn.

When the tool is assembled, the scriber, *b*, is set to zero, and fastened with a thumbscrew, *e*. Then, using the standard length gages, 1, 2, 3, 4, 5 and 6 inches long to get the positions, six holes are drilled and reamed through the slide and beam at the same setting. The same operation is performed when lapping, after the bushings are placed into the beam and slide. An important part of the tool is the threaded screw, *f*, the construction of which requires great accuracy. To make sure that the lead is perfect, when the last cut is taken go over the work three times with the tool in exactly the same position, using a magnifying glass to see whether the tool cuts any at either side of the thread; if not, it may be assumed that the lead is all right. The diameter of the screw is 3/8 inch, and it is cut with a lead of 1-20 inch. At the bottom end of the screw is fitted the piece *i*, which is hardened and ground. The curved part of *i* acts as the forward bearing of the screw, being seated in the 45-degree counterbore in the body, *a*. At its upper end is fitted the graduated barrel, *g*, and the speeder, *h*.



Parts of a Six-inch Micrometer Surface Gage.

Referring to the details, the movable part, *a*, of the instrument is made of tool steel. The two holes in the body for the movable scriber, *b*, and the beam, *l*, should be bored at one setting on the angle-plate, in order to secure perfect parallelism, which, of course, is necessary for accurate measurements. The hole for the scriber should be 1-64 inch large, but the hole for the beam should be a perfect sliding fit. The bottom of the hole for the scriber is closed by a piece, *n*, screwed in. This piece is bored out with a 45-degree counterbore to form a bearing for the lower end of the screw, and is hardened and lapped in the bearing. The threaded part is 11-16 inch diameter, 40 threads per inch. A slot 1/8 inch wide begins 5-16 inch from each end, and is carried through into the wall on the opposite side of the hole as indicated by dotted lines. In these grooves slides the movable scriber, *b*. There are forty divisions along the side of this slot 0.025 inch apart, these giving the tool a range of one inch with micrometer measurements between plug settings. In the beam hole is cut a slot 5-32

The sliding scriber, *b*, is made of tool steel hardened, ground, and lapped on the point, and combined with it is the micrometer nut, which part is drawn to a spring temper. This nut is split and adjusted by two small screws to compensate for wear. On the scriber is the zero mark, which forms the datum line from which the measurement is taken. To make a neat appearance, the cap, *k*, is placed on the bottom end of *a*, where it is held in position by being made to fit tightly to the bored recess. The setting of the tool is accomplished by loosening the speeder, *h*, and turning the barrel on the screw. When the adjustment is made, the speeder is again tightened down, thus locking the screw and the barrel together. The beam, *l*, is screwed into the base block, *m*, down to a shoulder. One of the most particular points to be observed when building a gage of this design is not to have the bushing holes in perpendicular alignment. With this precaution the plug, *d*, will enter no other hole than the corresponding holes in the beam and head for which it is intended. L. NORDEN.

INSERTED BLADE CUTTER CONSTRUCTION.

Editor MACHINERY:

An article in the April issue of MACHINERY about inserted blade milling cutters attracted my attention and reminded me of an inserted blade milling cutter which I saw in the exhibit of the Pratt & Whitney Co. at St. Louis last summer. In my opinion the method used for securing the blades to the body in this particular cutter was the simplest and most effective that I have run across. As the construction of these cutters may not be familiar to many of your readers, I hereby submit a description and explanatory sketches of same.

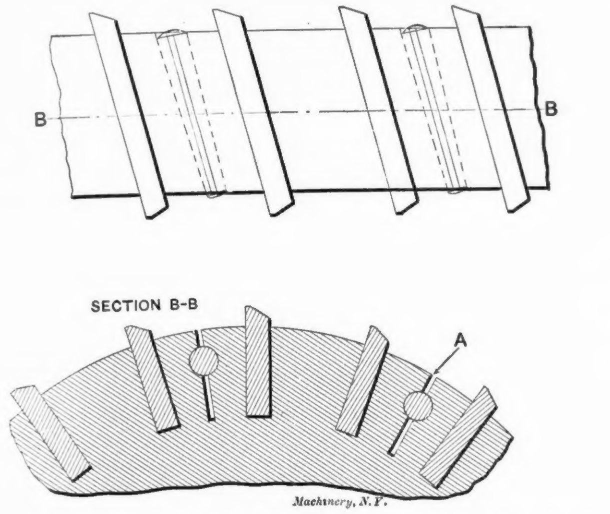


Fig. 1. Pratt & Whitney Method of Securing Blades to Body of Axial Cutter.

Two kinds of cutters were exhibited, one with blades set into the body on an angle with the axis of the cutter, and another with the blades parallel with the axis or center line. The former was to be used for face milling; the latter for end milling. The blades in both, however, were secured to the body by the same method.

As will be seen from the cut, the blades are set into rectangular slots in the body and held in position by means of taper pins which wedge the metal of the body firmly against the sides of the blades. There is only one taper pin for every other blade, the pin spreading the metal equally on each side of a narrow slot *A* located half way between the slots for the blades. My attention was called to the fact that the distances between the teeth must be such as to insure on the one hand perfect holding qualities (that is, the metal between the slot *A* and the slots

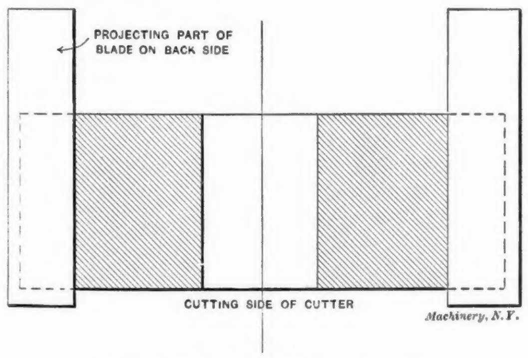


Fig. 2. Section of an End Milling Cutter.

for the blades must not be too heavy to allow good springing action when forced sideways by the taper pin), and on the other hand a strong and durable body.

On the cutter with the blades inserted on an angle the taper pins and the slots *A* were also, of course, on an angle, being parallel with the blades. This angle seemed to be about the same as the angle generally used on solid spiral fluted milling cutters, or about 15 degrees. Another feature that I observed on the cutters to be used for end milling was that the blades were projected quite a considerable amount on the back side, allowing for adjustment when the cutting faces of the blades were worn down near to the body. H. D.

AN EXPANSION CHUCK.

Editor MACHINERY:

In the March issue there was shown an expansion chuck which I know to be a very good tool, and as its similarity is so close to the one I designed about a year ago, I thought it might be of interest to some to show same. The work it was designed to hold was the very same as described in the aforesaid article, *i. e.*, brass-lined cast iron cylinders.

As will be seen, the jaws, *F*, are operated by the handles, *G*, which are fast on the cam ring, *B*; the cam ring in turn is held in position by the drive ring, *C*. This ring also serves to close the slot in the body of chuck through which the ⅞-inch square piece, *E*, transfers the cam motion to the centerpiece, *D*, which in turn acts upon the jaws, thus expanding same in cylinder. A coil spring, *M*, is placed at the lower end of each jaw, which is covered by a cap, *J*, and a circular spring, *L*, engages in the

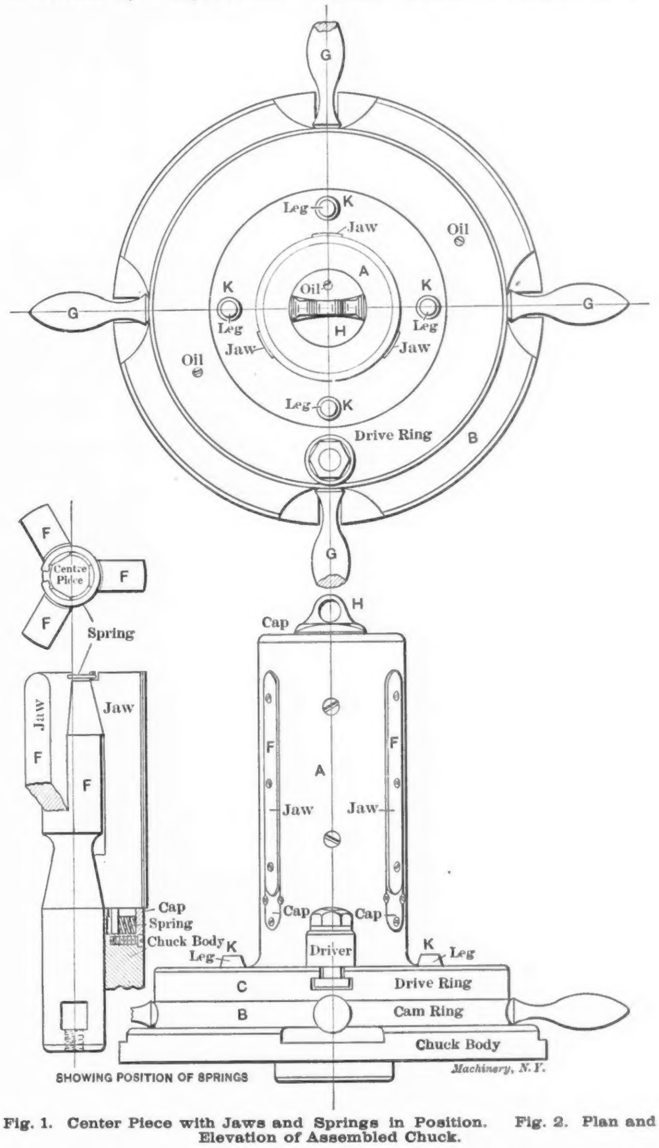


Fig. 1. Center Piece with Jaws and Springs in Position. Fig. 2. Plan and Elevation of Assembled Chuck.

three jaws on the top, which serves to contract them when the cam ring is released. The tapped holes in the jaws are used when cylinders of a different size are to be faced as adapting jaws are placed in position. Fig. 2 is a drawing of the assembled chuck. Fig. 1 shows the centerpiece with jaw and springs in position, and Fig. 3 the parts in detail. The cap on top of the chuck serves to close the opening and as a hold for the hoisting hook, so that the chuck can easily be lifted on and off the machine. J. M. STABEL.

Rochester, N. Y.

SOMETHING ON THE HEAT QUESTION AS AFFECTING THE ACCURACY OF MACHINE OPERATIONS.

Editor MACHINERY:

Our meetings were hardly of the A. S. M. E. class, but they were generally entertaining and often instructive. There was never any formal notice of a gathering, but sometimes after

dinner as we sat talking and smoking in the dining room attached to the works, one of us would introduce a subject either from his own experience or culled from the press. Then an easy flow of criticism or recital of experiences would follow till the bell event, to be continued if unfinished on the next day. Once a subject was mentioned, we generally managed to keep the conversation rolling for a while; we had most difficulty in finding new things to talk about. As may be expected, there were not many of us. Most of our fellow diners had enough—or too much—work in working hours to worry themselves with it in their spare time.

At one of our meetings the subject of keeping work of a precise nature at one temperature was discussed. It was introduced by a bluff old fellow, who though a thorough mechanic, had a slight disdain for things scientific. He read a paragraph out of a paper describing how a screw-cutting lathe at the British national laboratory was installed. It seems

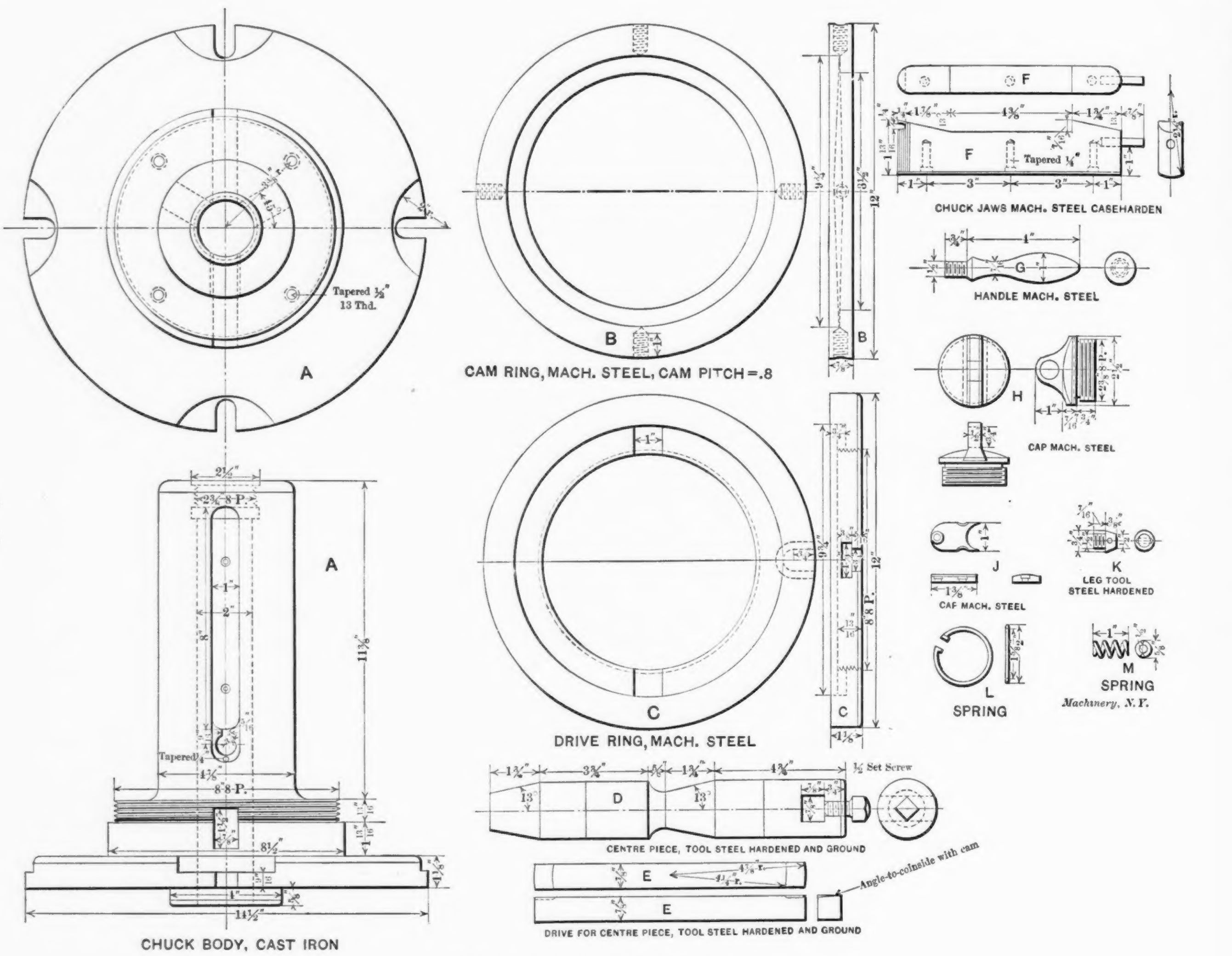


Fig. 3. Parts of Expansion Chuck.

that the lathe house proper is made of glass and is air tight. It is entirely surrounded by an outer house, the intervening space being filled with the heating and driving apparatus. Hot air is used to control the temperature and the lathe is controlled by electric switches in the outer house. No one is allowed in the inner room when the lathe is in use lest the heat of his body should affect the work in progress. This, he contended, was absurd, as many other things would affect the result in quite as great a degree as the heat of a person's body. The heat of the spindle bearings, the heat generated in the work itself, or the wear of the tool would all put in errors, even if the accuracy of the lathe guide screw was up to it, which he doubted. There was not much discussion on this as no one had any experience anything approaching to this in accuracy, but the majority who had anything to say seemed to be inclined to agree with him. However, it drew forth an experience interesting enough to present.

The relater was at the time it took place charge hand over a number of machines, amongst which was a milling machine, whose operation was to finish mill a thin rod to length. The rod passed between two side mills, whose distance apart determined its length. The error permitted was very small. The rod was about 8 inches long and had a fair amount of milling on it before this stage. When the job was started he was worried by the number of wasters returned by the inspectors, short in length. The error ranging from 2 to 4 thousandths. It was not an occasional lot that was wrong, but out of the majority of lots that went up, three or four were returned. First thoughts were that the cutters might be loose on the mandrel, but that was very soon dismissed. Then he thought that the work was not put in the jig properly, the rods projecting too far one way, which would take too much off one end without touching the other. But on testing it was found that too much had been cut off *both* ends. This was a problem. However, while watching the machine in operation shortly afterwards, the cause suddenly struck him. The lubricant used was a very thick oil and to remove this before the final operation, and the cuttings which clung to the work, he had installed an open kettle, filled with water and soda, heated with a small gas stove. The rods were put into perforated cans, dipped into the kettle, and after the can was given a shake the rods would be fairly clean. Upon looking into the kettle he found the water nearly at boiling point, and hung inside was a can full of rods evidently waiting for the operator. He hovered around awhile and saw the operator take the can out, and with the aid of some waste catch hold of a rod, clap it in the jig, tighten up and start the feed. Taking the coefficient of linear expansion of the metal at 0.000007 and the increase in temperature about 90 or 100 degrees, the expansion in the rod would more than cover the discrepancy between the rod and its gage. The temperature of the water was lowered, and it was pointed out to the operator the evil of milling the rods while hot; afterwards he had no trouble.

McAnic.

SOME PERSONAL REMINISCENCES.


Joseph H. Springer, Sr.

Editor MACHINERY:

I have read a number of articles in our trade papers from the pens of present-day mechanics that are misleading to the reader; as, for example, when the reader is told that the craft of pattern making was unknown forty years ago. This is a mistake, for even when the flour mills were built on the Brandywine River, James Rice had an iron and brass foundry in Wilmington, Del.; Malon Betts, of tne same town, employed pattern makers and molders one hundred years ago; and nearly as long ago Bush & Bonne made carwheels there. When I went as apprentice to Evan C. Stotsenburg in the spring of 1849, there were four large iron foundries in Wilmington. Stotsenburg's was the largest, employing, on an average, 8 to 10 patternmakers and 50 to 70 molders. Every man in the pattern shop had served a five years' apprenticeship, and had papers to show for it. The journeymen in the foundry were all good molders, able to make castings in green sand, dry sand, skin dried, or loam. We had in the foundry several steam jib cranes, two large core ovens, two large cupolas, and other tools. The cranes and cupolas were equal to any that are in use to-day in the most modern plant. The blast for the cupola was from a pair of twin blowing engines, and molders did not stop working when the blast went on, but gave a helping hand to any shopmate who was a little behind. Molders' and machinists' wages between 1850 and 1860 ranged from $10.00 to $12.00 per week, patternmakers receiving about 10 per cent. more. I say without fear of contradiction that mechanics did a fifth more work during the years I have named than they do at the present day, notwithstanding the many improvements in machinery and the present higher wages. Many of my readers will say: "We do better work now than did those old 'has-beens,'" but your work is not even as good.

In those days an apprentice was bound to his employer, and in many cases had to live with his master during the term of his apprenticeship. I served two-and-one-half years in the pattern shop, and the same length of time in the machine shop, receiving $2.50 per week from first to last. In common with other apprentices, I had to open and close the shop before and after working hours. For six months my work was cutting bolts with stock and dies. We had only one power bolt cutter, run by the engineer, who also attended to his engine and did his own firing, for which he received $1.25 per day of about 12 hours. The tools in the machine shop were very crude. Most of the lathes had endless chain feed, and the planers had a screw the full length of the bed, working through a nut bolted to the under side of the table. Drill presses had no power feed. Taps were made on a small lathe with hook tool and chaser in the hands of a skillful mechanic. In addition to this foundry, Wilmington had two machine shops. Harlins & Hollingsworth, the larger, employed from 800 to 1,000 men. They built passenger cars and steamboats, and had a large business in their machine shop.

In the spring of 1858, I moved to Philadelphia, Pa., and went to Wm. Sellers & Co. as a patternmaker, receiving $14.00 per week. It is not out of place to say here that the man who worked at the next bench to me, Mr. J. K. Jones, is still in the employ of the firm as foreman of the pattern shop, a position he has held for forty years. There were then about 250 men in the machine shop, 25 in the blacksmith shop, and 100 in the foundry, of whom 18 to 20 were patternmakers, a total of nearly 400 men besides the office and drafting room force. One man took all the time, booked it, and made out the payroll. According to the rule then followed in both Wilmington and Philadelphia, the men were paid in gold and silver on Saturday after working hours, up to the preceding Friday night. All was harmony between men and master, there was no red tape and no striking.

When I entered the employ of Wm. Sellers & Co., I found a shop system that is unequaled at the present day in any plant. When a machinist was employed, he reported to the storeroom for tools. If he was a vise hand, he told the storekeeper his vise number. The storekeeper went to his vise, unlocked his drawer, checked up his tools to make sure that they were in good order, and the man receipted for them. When a tool was broken or worn out, he returned it to the storekeeper and secured a new one for it. This rule held all through the shop. In the foundry a record of each molder's work was kept by the boss laborer in the cleanings shed, good castings being placed to his credit and bad castings charged against him, so that the firm could soon tell if he was a profitable man to employ. The flasks were all of iron, and a flask of any size could be made up at short notice.

I find very few all-round mechanics in the present-day machine shops; a man is either a lathe, planer, or boring-mill hand, and the same is the case in the foundry, a molder is a green sand, dry sand, or snap molder. If he is a green sand molder, he must have a pattern, flask, cope, and drag. If the foreman tells him to bed a pattern, he is lost, and there are only a few who know how to make a casting without a pattern. Loam molders are one in a thousand. If you find an all-round molder in a city foundry, you will find either a man who has served his time in some small town or one of the old "has-beens." I trust the time is not far distant when a boy will have the opportunity to serve his time at his trade as his father did. A few years ago I was in an Eastern machine shop that build a number of standard tools, such as small lathes, milling machines, etc. The gentleman who showed me through the plant, also showed me the stockroom. Here I saw the parts of the tools all stacked up after they had passed the milling machine. I said: "I presume these parts are now ready to be assembled." He smiled and said: "You know, Mr. Springer, we build very fine tools. These parts all go to a department where they are run through very accurate planers; they are then assembled and tested." This the milling machine and handyman, made from a laborer, cannot do, hence I found that even now manufacturers who build and turn out good work must at the end have some skilled mechanics.

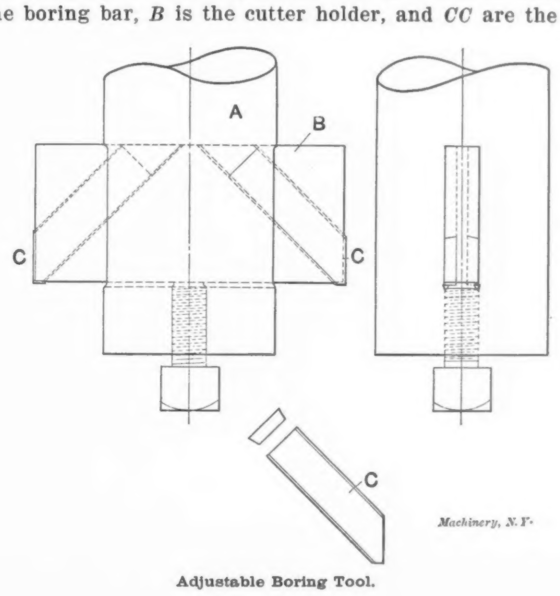
Louisville, Ky. JOSEPH SPRINGER, SR.

Joseph H. Springer, Sr., was born in Wilmington, Del., 1834. He is a lineal descendant of the original Swedish colonists in America, the American history of the family beginning with Carl Christopher Springer, of Stockholm, Sweden, who settled at Wilmington, Del., in 1646. Mr. Springer served a six years' apprenticeship with Evan Statsenberg, who had a foundry and machine shop in Wilmington; two and one-half years were served in the machine shop, two and one-half years in the pattern shop, and one year in the drawing room. At the expiration of his apprenticeship he was made general foreman, and this position he filled two years. He has since worked for William Sellers & Co., Bement & Dougherty, the Bethlehem Iron Co., the Edge Moor Iron Co., the Keystone Bridge Co., the Niles Tool Works, Fraser & Chalmers, and others, for which firms he has served in the capacity of foreman, draftsman, machinist, nechanical engineer, superintendent and owner. Mr. Springer has been prominently connected with interesting engineering work, particularly with the erection of the Kentucky River viaduct in 1876 to 1877, which demonstrated to American engineers for the first time, it is said, the important advantages offered by the cantilever bridge, in permitting erection without false works.

ADJUSTABLE BORING TOOL.

Editor MACHINERY:

The cut shows a new adjustable boring cutter in which *A* is the boring bar, *B* is the cutter holder, and *CC* are the two



Adjustable Boring Tool.

cutters of self-hardening steel. Two dovetail slots are milled out on opposite sides of the cutter holder at the angles shown. Into these dovetails are driven the self-hardening cutters *CC*. They are then ground off on the ends to suit the required diameter of hole. The advantage of this style of cutter lies in the fact that when the cutters have worn they can be easily driven out and reground to size. Experience shows that no fastening for the cutters is required since the angle at which they are set was chosen so that the natural tendency of the cutter to back out is eliminated. This cutter holder has been in use at the shops of the Alliance Machine Co., Alliance, O., for some months with most satisfactory results.

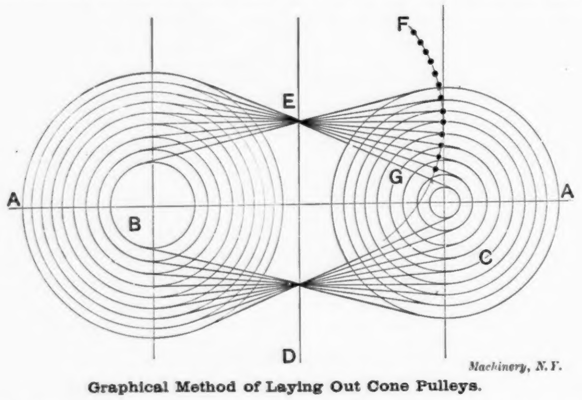
Alliance, O. F. A. HOBBS.

GRAPHICAL METHOD OF LAYING OUT CONE PULLEYS.

John Swanberg.

Editor MACHINERY:

My graphical method for laying out a pair of cone pulleys is as follows: First draw straight line *A A* supposed to connect the centers of the cones to be laid out; then set off the centers of the cones *B* and *C* on line *A A* (full size is best); then bisect the distance between the centers of the cones and draw another line *D E*. Now assume the size of the two cones—say the largest is 25 inches and the smallest 3 inches diameter. Then draw a line tangent to the circles, or the line representing the inside of the belt *G*, which will intersect the line *D E* at *E*, and taking the point *E* for a center scribe the circle *F*. Then divide the circle *F*, commencing at the line of the belt *G*, into as many parts as needed, of a length to suit the required speeds. Draw the other radiating belt lines through the point *E* and the divisions on the circle *F*, extending them toward the cone *B*, and they will be the in-



Graphical Method of Laying Out Cone Pulleys.

side of the other belt lines. Draw circles tangent to these lines and you will have all the diameters of the rest of the steps of the cone to match the first, and the belts will all correctly fit the steps. This is only an approximation rule, but it is much closer than other diagrams for the same purpose.

Pearl River, N. Y. JOHN SWANBERG.

SEEN IN A JOBBING SHOP.

Editor MACHINERY:

Among the many comical occurrences which I have witnessed in my wanderings as a journeyman, I will mention the following. An inventor (a would-be all-around mechanic) had occasion to drive a hardened dowel pin, and being in a hurry (as most inventors are), he failed to see a hammer near enough to suit his Royal Highness. He simply picked up my 6-inch hardened steel square, grasped it by the beam, and using it as a handle and the head as the hammer proper, he started in, but, needless to say, I did a little starting in on my own hook.

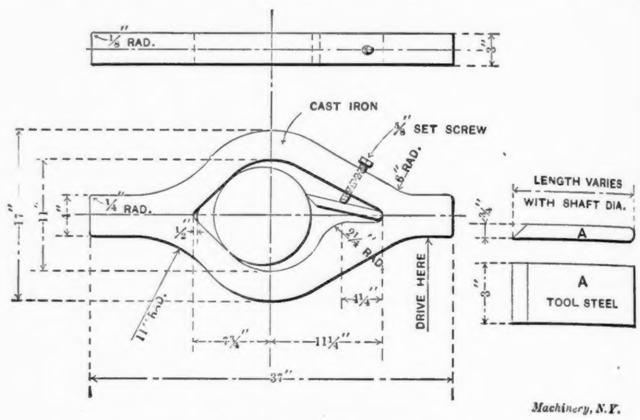
Two other "stunts" I witnessed which caused me to laugh good and plenty (because the tools "abused" weren't mine) were using a one-inch micrometer as a C-clamp, and a 3-inch vernier as a monkeywrench. I have also seen a 6-inch flexible steel scale used as a scriber. The above may sound quite "fishy," but they are facts nevertheless. All of the above were the acts of "inventors" who no doubt took things for granted. R. A. LACHMANN.

Chicago, Ill.

LATHE DOG FOR HEAVY CUTS.

Editor MACHINERY:

The cut shows a lathe dog which I have used for years in turning large forged shafts where heavy cuts are to be taken. I have taken cuts of 1½ inches without slip; in fact it positively will not slip. The cut will show the construction clearly



Lathe Dog for Heavy Cuts.

so that little explanation is necessary. The frame is made of cast iron and the driver *A* is of tool steel, made in different lengths to suit the work to be turned. J. MAYER.

Monongahela, Pa.

* * *

HOW AND WHY.

A DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST.

Give all details and name and address. The latter are for our own convenience and will not be published.

Editor MACHINERY:

Answering question No. 42, by H. F. S., in "How and Why" column, I would suggest that the inquirer use brass forms instead of iron, as the acids used in tanning discolor the leather when brought in contact with iron; in dampening or wetting the leather be sure and change the water frequently, or better still, use running water. The roughness he complains of depends on the firmness or quality of the stock used and cannot be avoided altogether unless the crimps are formed or pressed with the smooth or hair side out.

Seneca Falls, N. Y. JAMES MALTBY.

47. H. J. R.—Please give me a receipt for tempering circular saws used in cutting metals. The saws are made from 4 inches to 6 inches in diameter and about ⅛ inch thick. How can the warping of saws be prevented or how can they be straightened after warping?

A.—We refer this to our subscribers. Our inquirer, however, will find under shop kinks in the May number, a suggestion for hardening thin saws without warping, also in "The Treatment of Steel" by E. R. Markham, published by The Derry-Collard Co., 256 Broadway, New York, some practical advice on hardening and tempering saws. In "Saw Filing and the Management of Saws," by Robert Grimshaw, published by Norman W. Henley & Co., 132 Nassau St., New York, there will be found information in regard to hammering saws.

48. L. J. H.—I propose to construct a train of helical gearing in which a driving gear, an intermediate gear and pinion and a high-speed pinion will be used, as shown in Fig. 1. The intermediate gear and high-speed pinion are cut 20 pitch and at an angle of 32 degrees. The question is: To what degree angle should the driving gear, *D*, and the intermediate pinion, *C*, be cut in order to exactly counterbalance the lateral thrust of gears *A* and *B*?

John Swanberg was born in Hernösand, Sweden, in 1863. He served an apprenticeship in the shipyard of his native town, and has since been employed in this country by the A. B. See Mfg. Co., Yale & Towne Mfg. Co., C. W. Hunt Co., American Brake Shoe & Foundry Co., and the Dexter Folder Co., with which concern he is now identified. His forte is ornamental pattern work for castings of all descriptions.

Answered by Mr. E. H. Fish.

A.—A spiral gear acts as a wedge; in Fig. 2 we have a wedge shown more acute than our gear, to be sure, but acting on the same principle. If we push the wedge with a force M, it will spread apart any two pieces between which it may be placed, with forces A and B acting at right angles to the two faces. These forces have to bear a certain relation to each other known as the parallelogram of forces shown in Fig. 3 which is obtained by laying off M in some convenient scale, say 1 inch for each pound, laying off A and B from O in directions perpendicular to the surfaces of the wedge and making them long enough so that they will make sides of a parallelogram as shown. Then, if A and B are measured with the

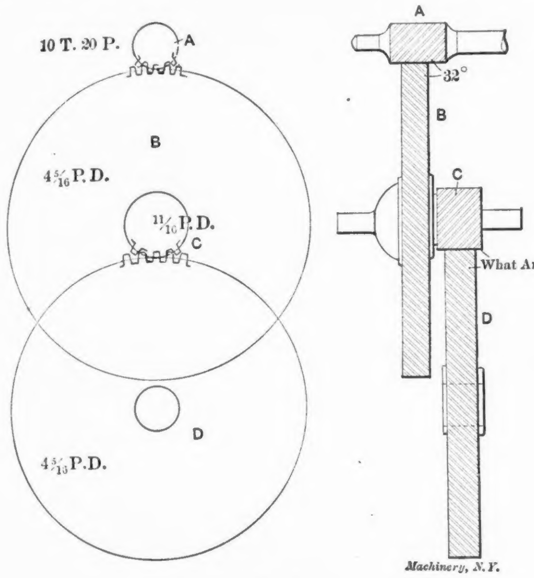


Fig. 1.

same scale with which M was laid off, we will find the pressure required. A corresponds to the force acting lengthwise of the shaft if the wedge is made of the same angle as the teeth of the gear. As gear tooth angles are usually measured, they correspond rather to the angles marked α in Figs. 2 and 3 than to the angle of the point of the wedge. From Fig. 3 we see that force $A = M \tan \alpha$. In the case cited, the force M is the pressure acting at the pitch line at right angles to the shaft. It would appear from the above equation necessary to know this force in order to get force A. But we really do not need to know what A is so long as we only have to make it equal to a similar force, A_1, acting in the opposite direction along the shaft. For the other gear we would have

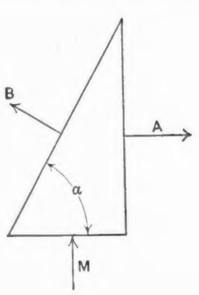


Fig. 2.

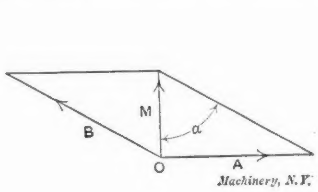


Fig. 3.

a similar equation, $A_1 = M_1 \tan \theta$, where M_1 is the force acting on the pitch line of the gear at right angles to the shaft and θ is the angle of the teeth on gear C. Since A and A_1 must be equal, we have $M \tan \alpha = M_1 \tan \theta$ where we know α to be 32 degrees, but do not know either M or M_1. But leaving out friction we know that the pressures on the pitch line of B and C will be in inverse ratio to their pitch radii or diameters, which in this case is 11-16 inches to 4 5-16 inches or 11 to 69. Changing the last equation to read $\tan \theta = \frac{M}{M_1} \tan \alpha$, and substituting 11/69, for $\frac{M}{M_1}$, and .625 for $\tan \alpha$, we get $\tan \theta = 0.1$, which corresponds to an angle of $5\frac{3}{4}$ degrees.

J. F. D.—Kindly inform me how to temper cold chisels to chip rough edges on chilled-iron castings. Also what shape of point or edge to use.

Answered by Mr E. R. Markham.

A.—The chilled iron castings referred to I take to be castings whose thin projecting surfaces are hard. Where there is no danger of breaking down into the portion of the iron we wish to save I have had excellent results from grinding the chisel perfectly flat on end as shown in Fig. 1, thus breaking the projections off rather than attempting to cut them.

If, however, it is necessary to make the chisel of a form that will cut, I have had best results with the form shown in Fig. 2, where the surfaces that make the cutting edge are rounding instead of flat, as is usually the case, and the included angle of these surfaces instead of being about 70 degrees, as is usually the case with a flat chisel used for cutting ordinary grades of stock, is nearer 85 degrees if it is possible to state an angle where the surfaces are rounded as shown.

However, the important part of making a chisel which must stand rough usage is the forging and hardening. When forging chisels for work where extreme toughness is desired, I am an advocate of the use of resin, dipping the heated chisel

FIG. 1

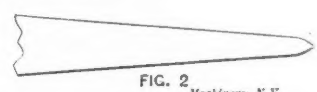


FIG. 2

in powered resin before the final hammering to shape. Hammer with light blows as the steel cools. Do not use any high heats; just *good forging* heats for tool steels.

After forging, I believe in re-heating to a low red and then laying one side till the red has disappeared, when it may be re-heated to a low red and quenched for hardening. Do not use *cold* water for this, as the steel is made too brittle by so doing, and is no harder than if hardened in water which has been heated somewhat—from 60 to 100 degrees F.

When dipping in the water it is a good plan to dip somewhat deeper than the point we desire to harden to, then gradually raise the chisel to the desired point; doing this we do not have any line where the metal is hard on one side and unhardened on the other.

When drawing the temper, check in oil rather than water when the desired color is obtained; the exact color cannot be stated arbitrarily. A deep brown with red spots will probably prove all right, provided proper heats were observed when forging and hardening; if heats too high in temperature were employed it will be necessary to draw the temper more.

When I wish a very tough chisel I prefer the following solution as a hardening bath rather than water, but care must be exercised in its use as it is a deadly poison: To six quarts of *soft* water add one ounce of corrosive sublimate and two handfuls of common table salt. When dissolved it is ready for use. Tools hardened in this bath do not require the temper drawn as much as if hardened in water; and tools hardened in water that is warmed somewhat do not need drawing as much as if hardened in cold water, as they are not as brittle.

* * *

A correspondent of the New York *Times* suggests the not entirely novel idea of utilizing the power of Niagara Falls at night and letting the waters flow in their pristine grandeur during the daylight hours. In other words, he would utilize the Falls to their full power during the night and give the tourist his money's worth during the day. But what the ultimate fate of Niagara Falls will be is a problem; the commercial tendency of the age will apparently make it almost impossible to defend this gigantic water power from the greed of corporate interests, which will be immensely benefited by its exploitation. A conservative estimate of the value of the power represented by the water falling at Niagara is not less than $100,000,000 yearly, and this figure means the rate of only $15 per horsepower year, or less than one-half the conservative rating. The fact to be remembered is that to whatever extent Niagara is exploited, it by right belongs to the whole people, and any benefit to be derived from its immense power should accrue to them, and not to the great profit of corporate interests. In other words, the time of giving away franchises of great value, and in perpetuity at that, has passed.

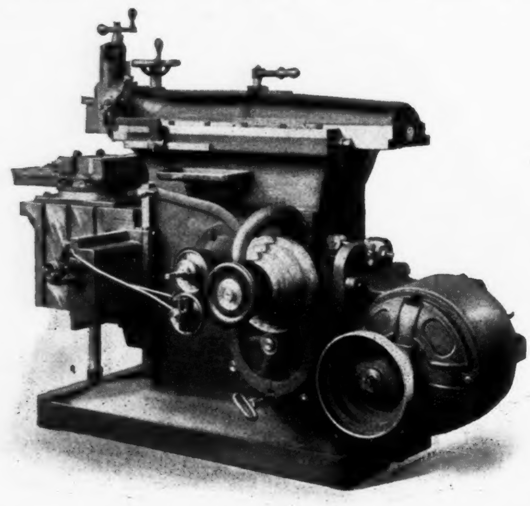
MACHINERY AND TOOLS.

A MONTHLY RECORD OF NEW APPLIANCES FOR THE SHOP.

TWENTY-FOUR-INCH CRANK SHAPER WITH MOTOR DRIVE.

The Cincinnati Shaper Company, Elam Street and Garrard Avenue, Cincinnati, Ohio, have brought out a 24-inch back-geared crank shaper, and one of these tools, having a motor drive as shown in the illustration given below, is now on exhibit at the Liege Exhibition, Belgium.

In the motor-driven machine the initial shaft, or shaft immediately below the four-step cone on the machine in the illustration, is driven from the motor by means of a pinion on the cone shaft meshing with a large gear on the initial shaft. In the belt-driven machine the cone pulleys are mounted on the initial shaft direct. In the machine illustrated the motor is of the constant speed type. The small wooden hand wheel on



Twenty-four-inch Motor-driven Crank Shaper.

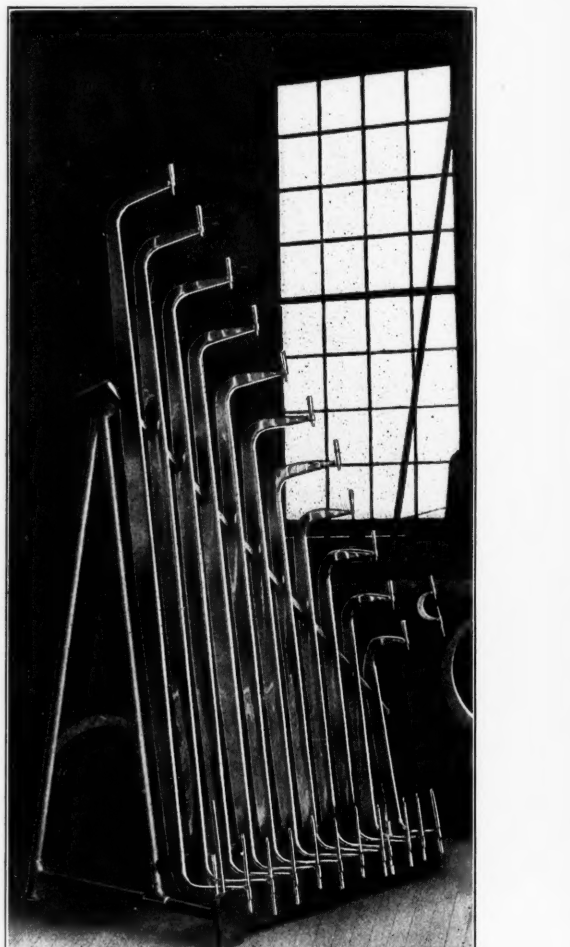
the cone shaft is for adjustment of the ram by hand, and the rod immediately below the large gear wheel is for changing the back gears in the machine, while the curved handle at the side of the machine operates a brake on the inside of the cone pulley. The leaf carrying the motor is hinged at its lower end, and is adjustable for the purpose of tightening the belt through the cap and setscrews shown.

The shaper is of rigid construction and is built by a system of jigs, insuring accuracy. The cross traverse screw is provided with a graduated collar reading to .001 inch and is also provided with a variable automatic feed, all feeds being obtainable while the machine is running. The head swivels to any angle and is graduated, and the down feed screw is provided with a graduated collar also reading to .001 inch. Full length taper gibs, adjustable endwise by single screws, are provided throughout. The ratio of gearing for the machine when single geared is 6 revolutions of the cone shaft to one stroke of the ram and when back-geared is 26 to 1. The outer support as shown is regularly furnished with each machine; this support is stiff and the surface on which the support rolls is truly parallel with the travel of the table. Ball bearings are provided under the elevating screw to the rail, this screw being of the telescopic form. The vise has a graduated swiveling base. The length of stroke is changed from the working side of the machine, and its position by means of a hand wheel on top of the ram; these changes may be made while the machine is in motion. An opening through the column under the ram provides for key-seating shafts up to 4 inches in diameter. Power down feed, revolving table, tilting table, tilting top for table, cone arbors, concave attachment, mould makers' vises, motor drives, etc., are furnished at an additional cost.

DAVIS TUBULAR MICROMETERS.

Mr. Frank M. Davis, who for nine years was assistant shop superintendent at the Edw. P. Allis works, is now engaged in the manufacture of tubular micrometers at 220 Oregon St., Milwaukee, Wis., and a set of bar micrometers made by him for the Allis-Chalmers Co., Milwaukee, Wis., is shown in the accompanying illustration. These micrometers measure from 18 inches to 84 inches and are designed for measuring pistons, cylinder heads, gear blanks, and all work that can be measured across the diameter. These tools can also be made into inside micrometers for taking the interior dimensions of cylinders, etc., by reversing the micrometers and mandrels in the frames. Beside the bar micrometers the Davis Company also manufactures bow micrometers, several of the smaller sizes of these tools being shown on the wall in the background of the illustration. The bow caliper is for measuring cylindrical work, such as shafts. The bar micrometer has a special advantage in that it can be used for measuring cylinders while being bored as the calipers will reach around the boring bar.

The calipers are made in the form of oblong tapering tubes well brazed, this form possessing the advantages of stiffness with light weight. The mandrel is held in the frame by a clamp bolt by which the mandrel can be secured at approxi-



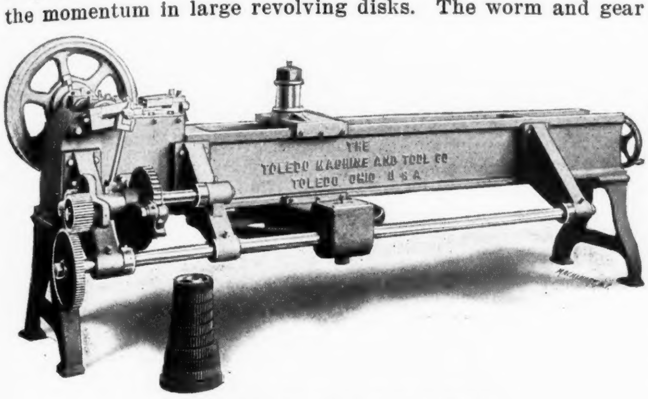
Set of Large Bar Micrometers.

mately the correct place for any given measurement. The micrometer head has a 1/16-inch movement in the frame, and by setting the micrometer at zero the micrometer head can be set to a wire end-measure gage by a nut on the end of the micrometer barrel. After the head is adjusted in this way it is clamped by a clamp bolt similar to that used for the mandrel. These micrometers are in effect comparators, being designed to always be set to standard end measures, this method preventing error through any possible springing to which large tools are liable.

The bow calipers are made in sizes from zero to 3 inches up to 72 inches, and the bar micrometers in sizes from 12 inches to 120 inches. Larger sizes will be made by the makers if wanted, and any of these tools will be sent to responsible firms upon request.

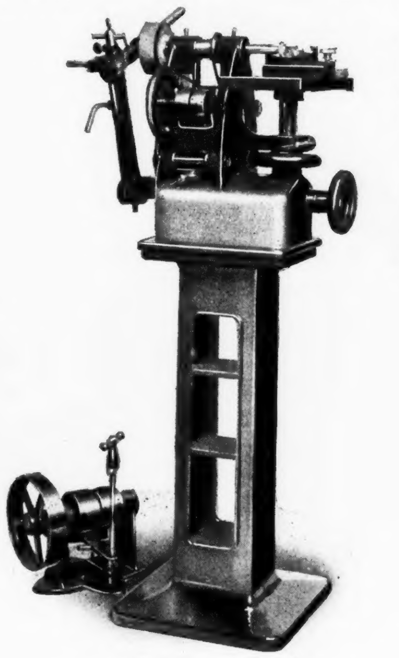
HORIZONTAL NOTCHING PRESS.

A horizontal press fitted with indexing or spacing adjustments for perforating metal rims is illustrated below. The press has a capacity for handling work 96 inches in diameter and for spacing or indexing divisions from 160 to 1,600 spaces. The indexing mechanism is fitted with a worm gear and worm, the makers claiming that this form of driving mechanism largely eliminates the inaccuracies which are often caused by the momentum in large revolving disks. The worm and gear



Notching Press for Steam Turbine Work.

are machined on special tools; the worm runs in an oil-tight box and is immersed in oil. The first driving shaft is fitted with a locking disk which also serves as a ratchet wheel. This wheel is positively locked at each stroke of the press or revolution of the crank shaft, immediately upon the release of the driving pawl. The change gears can be easily removed and various combinations made to obtain the different spacings required. Only one locking disk is furnished, but by means of the adjustable crank head connected by a rod to the locking disk plate, one or more notches may be used in spacing, thus reducing the number of change gears re-



Tap and Die Grinding Machine.

quired for making different divisions. So accurate is the indexing and spacing of notched work by this machine that it is not necessary to file or mill the disk slots in packing to obtain uniformly straight grooves even on reversing the disks. The machine has just been designed and built by the Toledo Machine & Tool Co., Toledo, O., for a special line of steam turbine work.

TAP AND DIE GRINDER.

The Greenfield Machine Co., Greenfield, Mass., have brought out a machine for grinding taps, die chasers, chucking reamers, surface work and light hand-grinding, generally. One view of this machine is given in the accompanying half-tone.

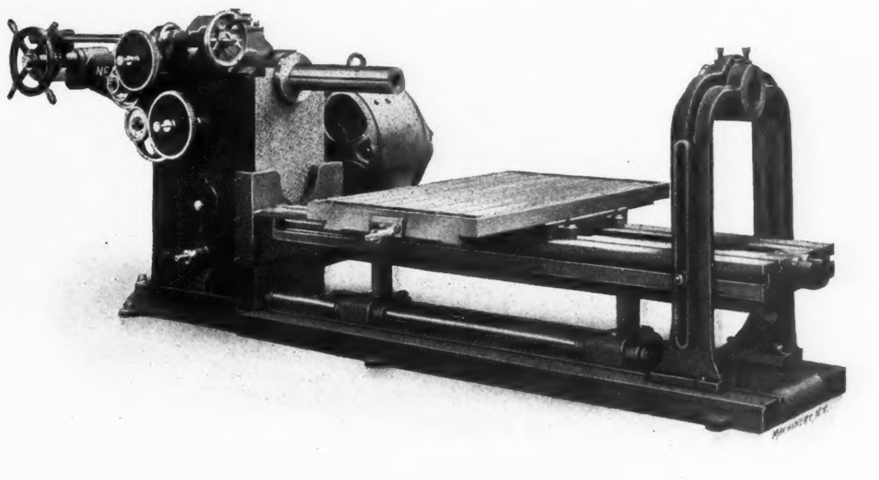
The spindle is fitted into cast iron boxes supplied with oil reservoirs and ring oilers. It has a 4-inch cup emery wheel mounted on one end of the spindle and a wheel holder for disk wheels on the other end. This holder can be easily removed and replaced by a ⅜-inch spring chuck for holding emery sticks, etc. Eight different speeds are provided to accommodate the different sizes of the wheels and there are means for tightening the belts. The table is elevated by screw and the horizontal hand-wheel shown, and when in position, can be clamped. The belt runs direct from a main line to the tight and loose pulleys on the machine, or a countershaft may be furnished.

The machine is fitted with a die-chaser holder and a tap-sharpening attachment. The chaser holder, shown to the right in the illustration, is provided with a gage against which the chasers may be clamped, thus permitting the whole set to be ground alike in relation to the thread, each having the same amount of clearance. This holder is mounted on a swivel base, thus allowing the chaser to be ground at any required angle. The holder is moved by hand across the face of the wheel, using the table guide and the knurled adjusting screw for feeding the work toward the wheel. The holder will grind chasers up to 2¾ inches wide, of any length.

The tap-sharpening attachment consists of two center heads mounted on a swinging arm which carries the tap across the face of the cup emery wheel as shown to the left in the view of the machine presented. A sleeve is provided on the swinging arm which may be clamped in any position, thus permitting the tap to be ground at any angle. The tap is indexed by means of a rigid tooth rest, and is held in place on centers, one of which is attached to a spring lever allowing the tap to be removed quickly. Means are also provided for holding a tap which has the ends broken off or one without center holes. The work is brought toward the wheel by means of the hand-wheel at the front of the machine. This fixture will hold all taps from ⅜-inch hand taps to 1½-inch machine taps, and with it chucking reamers can also be ground.

HORIZONTAL BORING AND DRILLING MACHINE.

The Newton Machine Tool Works, Philadelphia, Pa., have recently built a horizontal boring and drilling machine having a 5-inch diameter spindle. The machine is shown in the accompanying half-tone. Six changes of automatic feed are



Newton Horizontal Boring Machine.

provided, running in geometrical progression from .0072 inch to .2646 inch per revolution of spindle. The spindle has a movement of 42 inches, has hand adjustment and hand quick return, and is driven by a 10 horse-power variable speed motor through back gearing having a ratio of three to one, by a hardened steel worm and phosphor bronze wormwheel of steep lead, with a ratio of 11 to 1. The speed range of the machine is 4 to 1 through the motor and back gears.

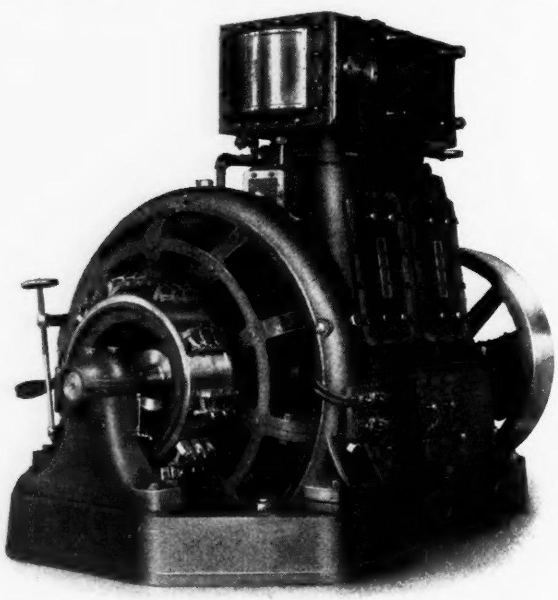
The knee is 26 inches wide x 9 feet long, and is fitted with a carriage 36 inches wide x 60 inches long with a cross movement of 36 inches. An automatic feed and stop may be fitted to the carriage where desired so that the machine may be used for various milling purposes. The elevating screws have both power and hand movements, and are 4 inches in diameter. The maximum distance from center of spindle to carriage is 26½ inches, and from the center of spindle to the knee 32½ inches.

STURTEVANT GENERATOR WITH CROSS COMPOUND ENGINE.

The B. F. Sturtevant Co., Boston, Mass., have perfected a line of small and medium sized generating sets in which the engine and the generator have been designed, the one for the other. In the April number of MACHINERY, we have described one of their generator sets of a type especially suitable for modern yachts and similar service, having a simple engine. Below we illustrate their latest form of generator set, having a vertical cross compound engine. The engine is entirely enclosed, has an oil-tight frame, centrifugal oil throwers, a water shed partition, and a forced lubrication by means of a direct driven oil pump, the oil pressure averaging 15 pounds. The specifications call for a regulation of 1½ per cent from no load to full load, by a Rites governor. The high pressure valve is of the balanced double ported type, with special eccentrically turned snap rings, with sliding joints, the low pressure valve being a flat balanced double ported sliding valve, arranged to lift from the seat. The cylinders are of close-grained, charcoal iron; the crankshaft is forged in one piece with a cast iron counterweight securely bolted thereto, and the main bearings are lined with Sturtevant white metal. All parts are made from templates and duplicates are carried in stock.

The specifications for the generator for this set are very similar to those given in the article referred to in the April number of MACHINERY.

The armature is of the iron-clad, two-circuit, form-wound, ventilated drum type. The armature is balanced both electrically and mechanically. The magnet frame is provided with a vertical adjustment for sizes of 16 K. W. and above. The field coils are shunt and series, separately wound and separately mounted on the pole pieces. The carbon brushes are



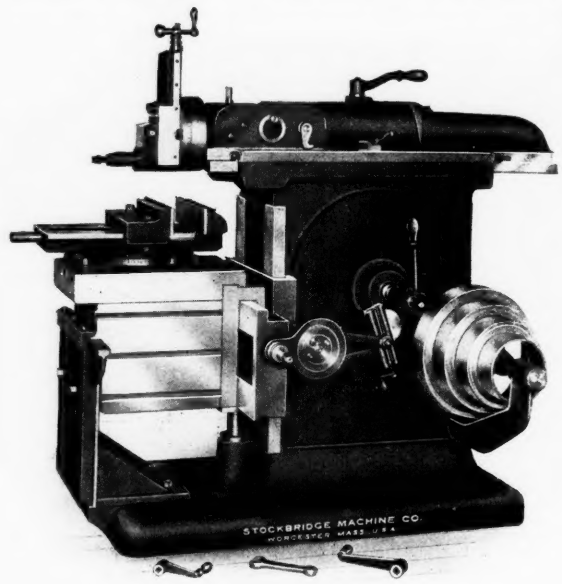
Generator with Vertical Cross Compound Engine.

separately removable and adjustable, and it is so arranged that the point of contact on the commutator remains the same as the brush wears away. The brush holders are staggered to even the wear on the commutator, and are mounted on studs projecting from the brush rigging attached directly to the magnet frame, this ring being arranged to be rotated for adjusting the brushes. The Sturtevant sets are built in a full line of both the vertical and horizontal types, ranging from 3 to 250 K. W. output. The engine and generator are set on the same base in the smaller sizes as in the illustration.

STOCKBRIDGE TWENTY-INCH SHAPER.

The 20-inch shaper recently designed by the Stockbridge Machine Co., Worcester, Mass., is strongly built to stand the usage of high-speed steel. The column is of box pattern, is extended 3 inches on top in front and gives a bearing surface for the ram of 32 inches. The ram is also of box pattern and is 46 inches long by 11 inches wide. An even cutting speed the entire length of the cut and a quick return of 4 to 1 are obtained by the use of the well-known Stockbridge two-piece crank motion, which is a feature of all Stockbridge shapers.

The machine has automatic feeds to both head and table, that are adjustable while the machine is in motion. The head has a graduated swivel which can be set at any angle and clamped in position by two bolts. The slide has a travel of 9 inches. The ratchet wheel for the vertical feed of the tool head is made of steel and its notches are cut to a fine pitch for power feed on hard material. The screw has a graduated collar reading to 64ths of an inch; the collar can be set to read from zero at all times without regard to the position of the screw. The table is of the box form with working surface on top 14 inches by 20 inches, and on the side 14 inches by 15 inches. The outboard bearing supports the table the entire width



Stockbridge Twenty-inch Shaper.

when in any position on the bar. The support engages with the table and is automatically raised and lowered by it. The table is raised sufficiently above the saddle to allow T bolts to be put in from the back as well as from the front. A cross feed of 26 inches is provided and is adjustable while the machine is in motion; it can be operated in either direction. The table hooks over the saddle as shown.

The driving gear is 20 inches in diameter and has a 3½-inch face. The shaper is back geared and has a four-step cone to carry a 3-inch belt. A telescopic elevating screw having a ball thrust bearing is used. The rocker arm is so made that a 4-inch shaft can be passed through the ram for key seating. The vise is of the swivel base pattern, has a graduated base and has steel faced jaws of 12 inches by 3 inches.

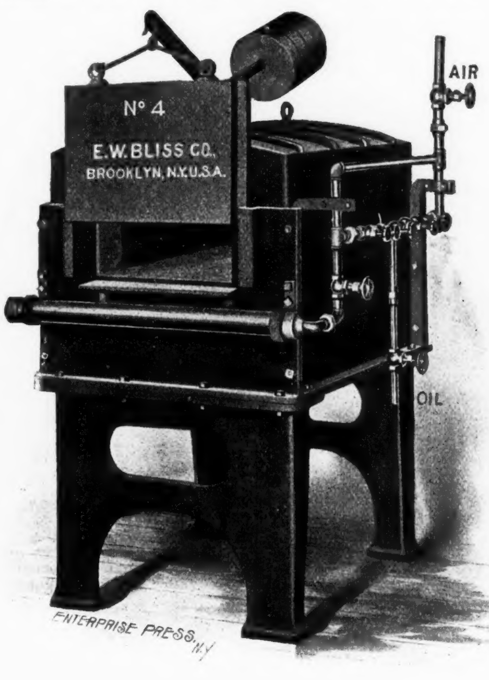
BLISS FUEL OIL-BURNING FURNACE.

The E. W. Bliss Co., 5 Adams Street, Brooklyn, N. Y., who, for a number of years past have made a specialty of building drop hammers, have recently added to their work a new department and are now preparing to equip complete drop forging plants. One of their new departures in this line is the oil-burning furnace intended for drop forging work, which is shown in the accompanying half-tone. Economy of oil was one of the chief considerations in the design of this furnace. The best service with it is obtained by using air as the atomizing agent under a pressure from 15 to 20 pounds. If tool steel is being forged a slow fire is necessary to avoid burning the stock. If soft steel or iron is used, however, the heat can be safely raised to a higher degree and an increased output thereby obtained. In small forging plants a good substitute for an air compressor installation, which is apt to be expensive, can be made by using a pressure blower giving air at about 1 pound pressure. The Bliss furnace may be used with either this or the higher pressure with only a difference in the burner.

A perforated pipe is placed directly under the opening in front of the furnace. The perforations in this pipe are small, running the entire length of the opening, and streams of air

are constantly flowing through the holes; this serves to keep the heat within the furnace and also helps protect the operator when he is removing hot bars or replacing them.

The furnace is made in four sizes. The opening and the interior of all these are of the smallest possible size, not only to avoid the wasteful use of oil, but at the same time to heat the material as quickly as possible. Oil as a fuel has the ad-

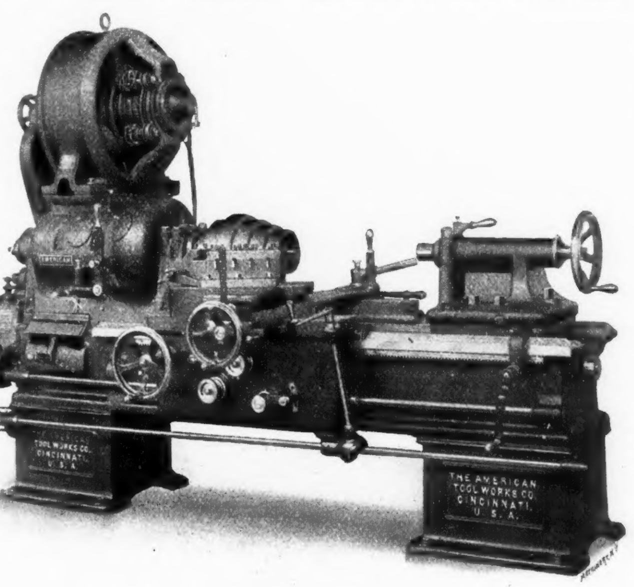


Bliss Fuel Oil-burning Furnace.

vantages over anthracite coal of greater cheapness and the absence of ashes and dirt in the shop. The heat generated by its burning may easily be kept uniform; another economy in its use is that when the forge is no longer wanted the closing of the valve instantly stops expense.

MOTOR-DRIVEN LATHE WITH GEARED HEAD.

We illustrate a 24-inch by 8-foot lathe with motor drive through an all-geared head; the machine shown also being equipped with a special multiple tool rest used in the rapid turning of cone pulleys. The motor is of 9 H. P., of the direct current variable speed type, of from 600 to 1,200 revolutions



Twenty-four-inch Lathe with Motor Drive.

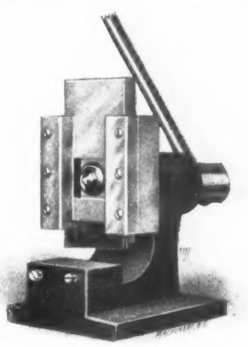
per minute. It is easily started, stopped and reversed by the controller handle placed at the right end of the carriage.

The speeds electrically obtained supplement the fundamental speed changes obtained mechanically through the all-geared head, which is a patented speed-changing device of the makers. This provides for spindle speeds, the ratios of which are 32.5 to 1, 10.8 to 1, 4.31 to 1 and 1.44 to 1. These speeds are obtained while the machine is in operation by means of the two levers shown on the front of the head, which operate positive clutches. Only six gears are used in this device, these being arranged to run at low pitch line velocities, thus reducing the noise incident to all gear drives. Adjustment for speed is determined by reference to the index plate under the front of the head.

This lathe is also equipped with a rapid change-gear mechanism, providing a wide range of changes for feed and screw cutting, each of which may be obtained while the machine is in operation, an index plate showing how to obtain any desired feed. The lathe, although motor driven, may be arranged for a belt drive by replacing the gear on the driving shaft by a pulley. The latter may be of wide face and large diameter, since the belt is not shifted to change the speed, which fact together with the high gear ratio, greatly adds to the power of the tool. The regular carriage equipment consists of a compound rest and a full swing rest. The lathe is manufactured by the American Tool Works Co., Cincinnati, Ohio.

PRECISION BENCH SHEARS.

Precision bench shears intended for use by die makers, lock makers, etc., are shown in the accompanying half-tone. The shear blade of this tool is 4 inches long, and the shear will split a 4-inch strip through the center; 3/16-inch sheet tool steel seems to be the limit of the capacity of the shear, although ¼-inch iron has been easily cut with it. Special care is bestowed on these shears during manufacture; the slides are all scraped to a surface plate, the adjustable gibs are finished bright, and the steel blades are ground true on their backs after hardening, so as to give a full bearing when drawn up against the slide by the screws. The motion of the shears is derived from an eccentric shaft, and a lever to be pulled toward the operator.

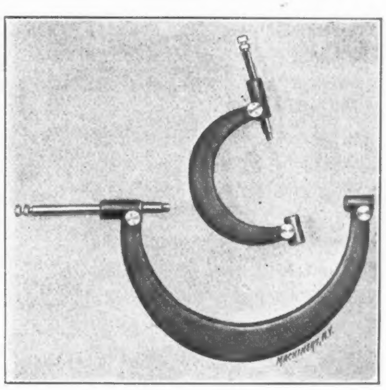


Precision Bench Shears.

The eccentric shaft pin is surrounded by a steel block which works in a slot in the upper slide. The chief considerations in the design of this tool were sufficient stiffness and accuracy in fitting. The shears are made by A. J. Machek, 401 19th Street, Milwaukee, Wis.

TUBULAR MICROMETER FRAMES.

A. J. Machek, 401 19th Street, Milwauke, Wis., is also manufacturing sets of tubular micrometer caliper frames, of which the 3-inch and the 6-inch frames are shown in the accompanying



Tubular Micrometer Frames.

cut. These caliper frames are made of high grade sheet steel, and are therefore light and stiff. They are intended to be used with the detached micrometer heads now being sold by several makers, one head being required for a set of frames to measure from 0 to 12 inches. The tubular frames are made especially for shops which are now equipped with a system of wire gages. Standards are not furnished regularly with the frames, but can be obtained by those desiring them. The adjustable anvil in the frame is held by friction. The grip, while amply suffi-

cient to hold the anvil when measuring, will slip if the tool should be mistaken for a clamp by the workman. The frames are made in four sizes at present, measuring from 0 to 3 inches, from 3 to 6 inches, from 6 to 9 inches, and from 9 to 12 inches. The frames, in connection with a detached micrometer head, make a set of micrometers of low cost.

GEARED SPEED MILLING MACHINE.

The cuts given below show two views of a positive geared speed milling machine recently put on the market by the Hendey Machine Co., Torrington, Conn., and a sectional view showing the speed and feed works of the machine. This miller differs from the standard machine in that it has a gear cone as shown, instead of a belt cone, on the main spindle for speed changes. With this positive drive a large range of spindle speeds is given, and the full range of feeds is quite independent of the spindle speeds. As shown in the figures, the driving shaft carries a large flanged pulley on the outer end, this pulley being belted either to an ordinary countershaft or to a motor as shown in Fig. 1. Sliding on the driving shaft, inside the base, is the regular form of bracket carrying the driving gear and intermediate gear, the latter not being shown in the cuts. This intermediate gear may be meshed into any one of the six gears forming the cone on the spindle, its position being controlled by the guiding handle fitted with a spring latch, this device engaging in the guiding slots and locking holes in the upper wall of the opening, as shown in Fig. 1. Six progressive direct spindle speeds are thus available, and as the machine is double back geared, a series of eighteen different progressive spindle speeds is supplied, having a range of 16 to 370 R. P. M.

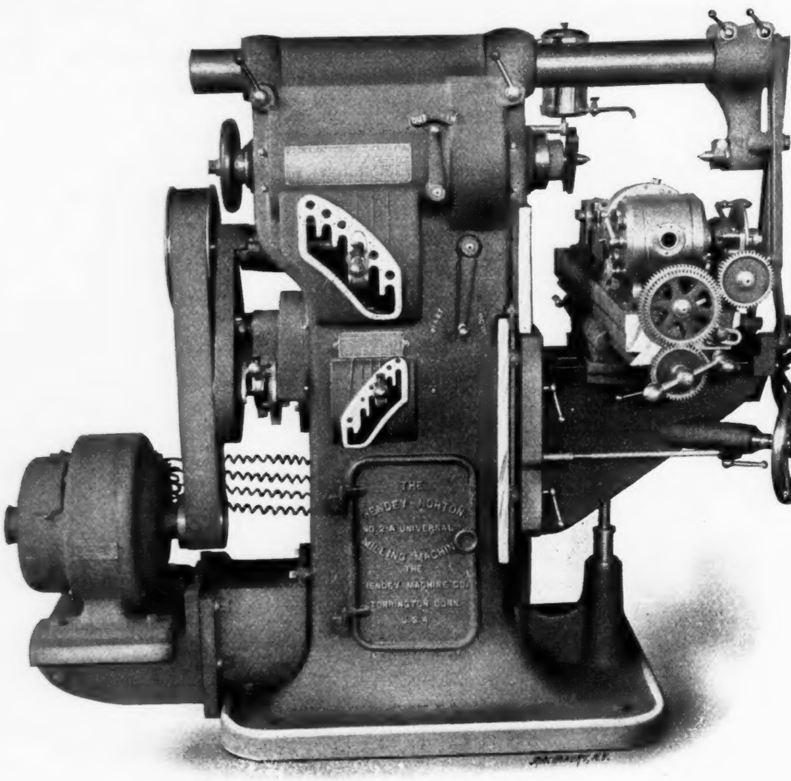


Fig. 1. Hendey-Norton, Geared Speed Miller, Left Side.

The back gearing may be constantly in mesh, if preferred, or it may be thrown out by means of the eccentric sleeve and lever, shown on the end of the back gear shaft in Fig. 2. When left in mesh it facilitates handling the entire range of speeds, as by means of the two positive clutches, one of which is mounted direct on the spindle and the other on the back gear shaft, one direct and two back gear speeds are handled for every setting made in the gear cone on the spindle. These clutches are controlled by the two levers shown on the left side of the machine, Fig. 1. All gearing of the above described combination, except the large face gear and large back gear, is made of steel. The clutches are of crucible steel and are hardened. The upper lever, shown in Fig. 1, controls the out and in clutch for giving either back gear or direct drive, while the lower lever controls the fast and slow back gear combination. The hand wheel attached to the rear end of the spindle furnishes a convenient means for rolling the spindle over by hand, either for entering gears or for setting a cutter close to the work.

The feed shaft is geared from the driving spindle with a chain and sprocket, as shown in Fig. 3, and the whole eighteen feed changes are available for each spindle speed. The feed index plate gives the table travel in inches running from ⅜-inch to 20 inches per minute. These speeds also practically apply to the saddle and knee as well as the table, as both are automatic. Any standard make of reversible motor may be used in place of a countershaft if desired, the method of mounting the motor being, as shown, very simple.

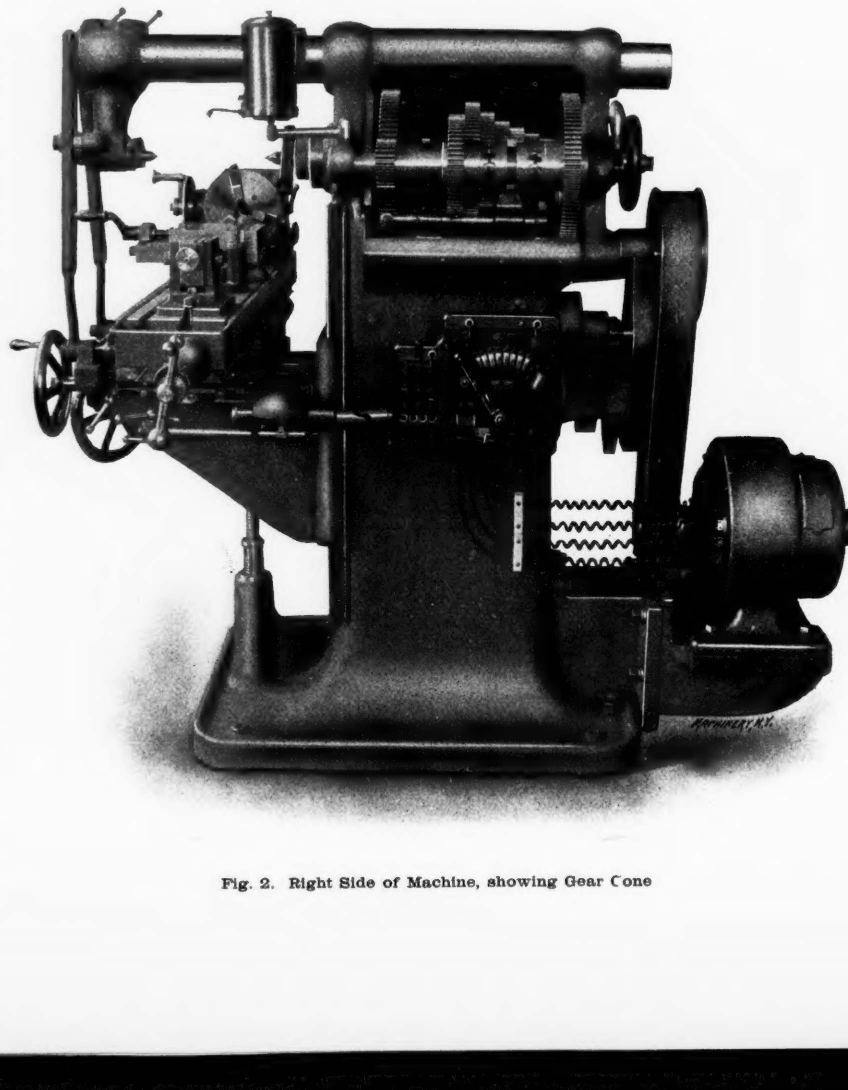


Fig. 2. Right Side of Machine, showing Gear Cone

A COLD SAW CUTTING-OFF MACHINE.

A newly designed and powerful cold saw cutting-off machine which has been recently shipped by the makers, the Espen-Lucas Machine Works, Philadelphia, Pa., to the Crocker-Wheeler Co., Ampere, N. J., is illustrated in the accompanying cut. This machine is similar in appearance to a bar cold

saw cutting off machine previously made by this company, but there is considerable difference in construction, the castings all being made heavier, steel replacing many iron castings. The drive is by a Crocker-Wheeler motor through a hammered crucible steel worm and phosphor bronze worm wheel and crucible steel gearing. The machine has a variable automatic feed, and automatic safety stop which throws out

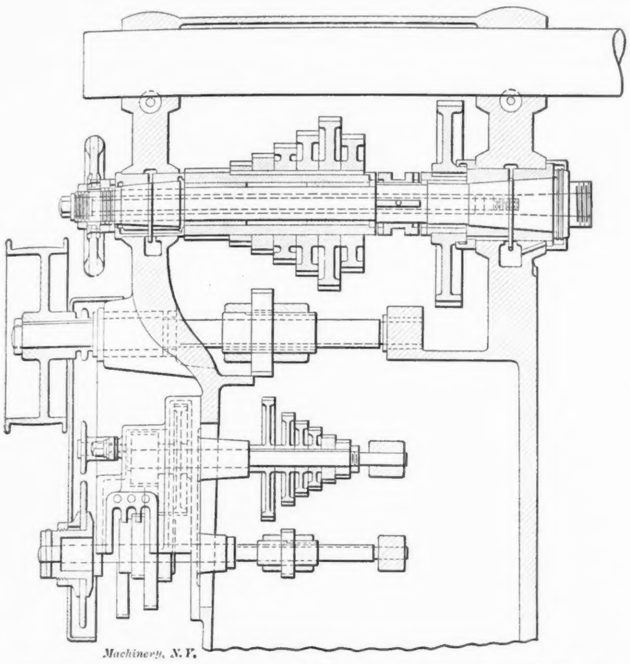
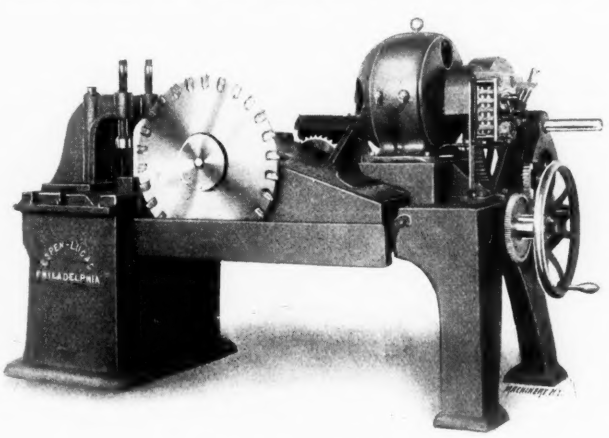


Fig. 3. Section through Speed and Feed Works. Geared Speed Miller.

the feed at any depth of cut. The clamps are swiveled and can be placed upon any part of the platen for cutting material either straight or at an angle, or they can be entirely removed for cutting parts of large odd shaped castings. This machine is capable of driving the high speed inserted tooth saw blade shown in cut to its greatest capacity. This machine has cut 6-inch steel bars of 25 point carbon in 6 minutes, and good results have been


Cold Saw Cutting-off Machine.

shown in cutting high carbon steel. In actual use, this machine has cut 175 pieces of 7½-inch 25-point carbon steel without regrinding the saw. These machines are furnished for belt or direct motor drive, and can, with slight changes, be arranged to saw rails, steel castings, forgings, etc., as well as bars.

DIAMOND POINT TOOL HOLDER.

The Elgin diamond point tool holder, illustrated herewith, is the result of several years of experimenting on the part of the makers, The Elgin Tool Works, of Elgin, Ill. The cutter is made of square bars of self-hardening steel, and is held at the correct cutting angle in the holder for sharpening. The cutter needs only to be ground on the top, so that the diamond shape is thereby preserved for a much longer time than otherwise. The holder is made of tool steel and is planed square on top and bottom. The cutter is held by an eccentric stud, no screws being used. The groove for the cutter is milled in the holder by gages.

Diamond Point Tool Holder.

The holder is tempered all over and all parts are interchangeable. The tool can take a cut the full width of the cutter, and can be used on either lathe or planer. The holder is made in two sizes, No. 1 being 1 inch by ½ inch, and the No. 2 holder 1¼ inch by ⅝ inch.

GRINDING AND BUFFING MOTORS.

The Lamb Electric Co., Grand Rapids, Mich., have put on the market a portable electrically driven grinder with emery and buffing wheels. We illustrate the quarter H. P. size of this apparatus. These motors are well balanced, insuring smooth running; shafts and bearings are extra large and long and the


Lamb Electrically-driven Grinder.

commutators are of well insulated copper. They are particularly designed for large torque or pulling power. The armature is of the slotted type, and the field magnets are steel castings of high permeability.

MISCELLANEOUS TOOLS AND APPLIANCES.

A screw plate recently brought out by the Conant & Donelson Co., Greenfield, Mass., has as its chief feature the reversibility of the die blocks, so that it may be used for either machine or hand work. The screw plate is made in sizes from 3-16 to 1½ inches.

The Reed Tool Co., Erie, Pa., are making a machinist's vise which may be swiveled in any direction, horizontal or vertical. Both horizontal and vertical swivels are provided with lock pins, but the friction clamps are sufficiently powerful to resist any ordinary force, the pins being merely to hold the vise in a fixed position.

A new machine for rapidly centering round stock is made by The Automatic Machine Co., Greenfield, Mass. The essential parts of the machine are a V-rest, adjustable to the diameter of the work, a bell-shaped center in the revolving spindle and a drill and countersink in the tail stock. Longitudinal slots divide the center into several prongs to give it a gripping action on the work when the latter is pressed into it.

A cutting-off machine for heavy work, embodying some new features of design, has been placed on the market by Nutter, Barnes & Co., Boston, Mass. The drive is of sufficient power for a 25-inch saw, 3-16 inch thick, cutting through metal of the maximum thickness allowed by a saw of this diameter. There are five changes of feed ranging from 1-5 to 1-2 inch per minute, and the carriage has a quick-return obtained by a clutch mechanism.

A quick-acting monkey wrench has been brought out by the Richards Tool Co., Boston, Mass. The quick motion of the jaw is obtained by rotating the handle of the wrench, which has an inner sleeve on which is cut a coarse-pitch helical groove. A projection extending down from the sliding jaw of the wrench fits in this groove and two or three turns of the handle cause the jaw to traverse through its whole range. A locking collar at the end of the handle, having small teeth fitting into corresponding teeth on the end of the sleeve, prevents the handle from turning, after the wrench is once set for a given size nut.

Furnaces for hardening, tempering and annealing for use with coal as fuel, where oil and gas fuels are not easily obtained, have been put on the market by the Kenworthy Engineering and Construction Co., Waterbury, Conn. The work is protected from the direct action of the flames by a grooved, tiled floor, the gas entering the heating chambers at the rear and passing out near the front. When the work is of such a nature that all contact with the flames must be prevented, a muffle is used for the protection of the work. The furnace is adapted for machine shops and other places where excessive radiation of heat would be undesirable, since the walls are well insulated. The furnace is built in three sizes.

* * *

FRESH FROM THE PRESS.

SUCTION GAS. By Oswald H. Haenssgen. Published by the Gas Engine Publishing Company, Cincinnati, O. 88 12mo. pages. Price, $1.00.

This is a monograph explaining the properties of suction gas, and the construction and operation of suction-gas producers, with practical information upon their use. It is a very brief treatise and one of the first to appear on this recent development of engineering.

ALTERNATING CURRENTS, THEIR GENERATION, DISTRIBUTION AND UTILIZATION. By Geo. T. Hanchett. .Published by John Wiley & Sons, New York. 175 12mo pages, illustrated. Price, $1.00.

There are to-day many practical engineers who have an inadequate idea of alternating currents and alternating-current machinery. Mr. Hanchett's volume is a very elementary tratise, free from mathematics, and designed to explain the basic principles of alternating currents in a way that may be understood by all. Readers who have had difficulty in grasping this branch of electrical engineering, through the perusal of textbooks now available, will find in this one just the help needed to give them an intelligent start and prepare them for the more advanced works, if they have sufficient knowledge of mathematics to master the latter.

THE LAY-OUT OF CORLISS VALVE GEARS. By Sanford A. Moss. Published by the D. Van Nostrand Co., 23 Murray Street, New York, as one of their science series. 108 pages, illustrated. Price, 50 cents.

The matter contained in this handbook first appeared in *Power* and *The American Machinist*, and was written to supply an adequate treatment of the design of Corliss valve gears which, strangely enough, has not been thoroughly covered in the standard textbooks upon valve gears. This little work of Mr. Moss' will be appreciated by draftsmen engaged in steam engine work.

STEAM AND STEAM ENGINES, INCLUDING TURBINES AND BOILERS. By Prof. Andrew Jamieson. Published in America by the J. B. Lippincott Co., Philadelphia. 780 12mo. pages, illustrated. Price, $3.00.

This is the 14th edition of Prof. Jamieson's well-known and popular work upon the steam engine, and in this edition he has added a brief chapter upon the turbine, setting forth some of the elementary principles of this type of engine. There have also been additions to several of the old chapters, others been rewritten, while the whole book has been recast, repaged, and a new index prepared. Some of the changes consist in additional matter upon pyrometers and calorimeters, superheated steam and Corliss engines. A feature of the book is the questions and answers at the end of each chapter taken from various examination papers, and these have been added to in the new edition.

NOTES ON HEAT AND STEAM. By Prof. Chas. H. Benjamin, Case School of Applied Science, Cleveland, O. Published by Chas. H. Holmes, 2303 Euclid Avenue, Cleveland, O. 93 12mo. pages, illustrated. Price, $1.25.

A new and revised edition of Prof. Benjamin's little treatise on thermodynamics, which has been favorably known as a convenient handbook for engineers and draftsmen for several years past. Many, who have felt compelled to consult elaborate works on this subect, have done so under protest and with the desire that something in the way of a condensed handbook were available. This work will be found to meet the situation exactly. It contains all the thermodynamic formulas that the practical engineer will be likely to need for his calculations upon heat and steam, together with some other matter not usually included in textbooks upon thermodynamics, such as data upon combustion and fuel, and information upon chimneys and their design. The new edition has a chapter upon air, gas and refrigeration cycles not in the earlier editions, and the other chapters have been added to and revised as needed.

MARINE ENGINES AND BOILERS, THEIR DESIGN AND CONSTRUCTION. By Dr. G. Bauer, Engineer-in-Chief of the Vulcan Works, Stettin. Translated from the German by E. M. and S. Bryan Donkin, and edited by Leslie S. Robinson. An English work, published in New York by the Norman W. Henley Publishing Co., 132 Nassau Street. 744 8vo. pages, illustrated. Price, $9.00.

This work was prepared by Dr. Bauer to supply the needs of young engineers, and he has succeeded in combining in condensed form both a great deal of practical information upon marine engineering and enough of theory to make the work valuable to the designing engineer. Dr. Bauer's unusual opportunity for securing practical data and drawings, and his extended experience as a marine designer, well qualified him for the work, and the book has been much appreciated in Germany and has had an extensive sale in that country. The translation into the English language has been made by well-known English engineers, and tables and mathematical work have been converted from metric units into English units. While there are several excellent works by English engineers which cover about the same field as this one, such as Seaton's Manual, students of marine engineering and engineers engaged in marine work will be glad of an authoritative volume embodying information about German practice. The rapid progress in Germany in the line of ship building and marine machinery during the past few years makes this German treatise all the more timely.

The book is divided into eight parts as follows: First, The Marine Engine, under which are treated cylinder dimensions and the utilization of steam; the arrangement of the engines, and engine details. Second, Pumps. Third. Shafting; The Resistance of Ships and Propellers. Fourth, Pipes and Connections, including flanges, valves, under-water fittings, steam and exhaust piping, feed water pipes, bilge pipes, circulating pipes, etc. Fifth, Steam Boilers, including descriptions of the different pipes and practical information upon firing and the generation of steam, forced draft, boiler fittings, etc. Sixth, Measuring Instruments. Seventh, Various Details. Eighth, Various Tables.

The book is handsomely bound and printed, is illustrated by a great many drawings and folding diagrams, and in selecting samples of marine practice from which the illustrations were made only the most modern types of marine engines and boilers have been taken.

TOOLS FOR ENGINEERS AND WOODWORKERS. By Joseph Horner. 340 pages 5¼ x 8 inches, and 456 illustrations. Published by Crosby, Lockwood & Son, London. Price, $3.50.

This work is by a well-known author who for many years has contributed articles of much technical merit to English and American trade papers, and who is the author of "Pattern Making," "Hoisting Machinery," etc. The book under review consists in a large part of selections from various articles contributed to the *English Mechanic* and the *Mechanical World*. The grouping together of matter upon iron working and woodworking seems ill-advised from the American point of view as the two trades are widely separated in this country, but such apparently is not the case in English practice. In the introductory chapter the author gives the distinction between a tool and a machine and briefly defines tools in general. Chapter I takes up the important subject of cutting angles and in it the author calls attention to the lack of uniformity, especially in metal working tools, and to the desirability of securing a more general agreement in shop practice. For instance, he says that there is little reason why the angle of clearance of metal cutting tools should ever exceed 5 or 6 degrees, but it often does without any apparent detriment to the permanence of the cutting edge. Under the head of The Chisel Group, Section 1, are five chapters: Chisels and allied Forms for Woodworkers; Planes; Hand Chisels and Allied Forms for Metal-Working; Chisel-like Tools for Cutting Metal by Planing, Etc.; The Shearing Action and Shearing Tools. Without further outlining the scope of the various chapters, we will refer to specific chapters such as that on saws, which seems to be of considerable value; milling cutters; boring tools for metal; hardening and tempering; standards of measurement; measuring tools; etc. In the review of a work of this character, of which a number have been brought out within the past year or two, one cannot help being struck with the fact that American small tools are about the only ones that seem to be considered worth describing. It would be very interesting to know to what extent the world is indebted to the American manufacturer of small tools for the development of the art, and to what extent he has drawn upon the tools and devices of European workmen for his models. In other words, whether the universal use of American small tools is altogether due to their excellence or whether it is due to the fact that America is about the only country in which small tools are manufactured and sold at prices that put them within the reach of the ordinary mechanic. The work is one that trade schools may be able to use profitably, but it contains little practical data for the journeyman. It reviews well-known tools and elementary principles, and the apprentice can doubtless read the book with considerable profit.

WEBSTER'S INTERNATIONAL DICTIONARY.—New edition containing 25,000 new words, and 5,000 illustrations. Published by the G. & C. Merriam Co., Springfield, Mass. Price, bound in sheep with marble edge, $10; with complete reference index, notched on edge, $10.75. Other bindings at various prices up to $18 for Turkey morocco with gilt edge.

Webster's Dictionary, beginning with the 1847 edition by Noah Webster, has had a long and honorable career (if a great work like this can be said to have a career) in the United States, and thousands of educated men and women think of it as the *only* authoritative reference for definition, derivation and pronunciation of the English language. The "International" edition is the successor of the once popular "Unabridged," being a thorough and complete revision of that work. The latest edition of the "International," recently issued, has 2,380 pages and includes 25,000 new words and phrases in a supplement. The pronouncing biographical dictionary and the pronouncing gazetteer or geographical dictionary of the world supplements have also been revised; the latter embraces the most recent statistics available. In short, the new edition may be truthfully said to comprise not only a defining and pronouncing dictionary but also a very comprehensive cyclopedia of information which scarcely any man can afford to be without.

NEW TRADE LITERATURE.

THE CROCKER-WHEELER CO., Ampere, N. J. A reminder of the warm weather that is to come is a pamphlet issued by this company describing motor-driven fans.

PETER A. FRASSE & CO., 92-94 Fulton St., New York. Price list of Boldi tool steel giving sizes and prices of steel and containing suggestions for the proper treatment of steel.

COLBURN MACHINE TOOL CO., Franklin, Pa. Illustrated circular of the new 30-inch vertical boring and turning mill made by this company, a description of which has recently appeared in MACHINERY.

DETROIT TWIST DRILL CO., Detroit, Mich. Catalogue of Graham twist drills and chucks, the former having grooved shanks to fit the jaws of the chuck. Also twist drills are listed of standard styles.

WESTINGHOUSE ELECTRIC & MFG. CO., Pittsburg, Pa. Illustrated pamphlet upon machine tool drives showing the application of motors to machine tools of different types and containing descriptive matter and tables upon the subject of motor driving.

P. BLAISDELL & CO., Worcester, Mass. Illustrated catalogue of engine lathes in sizes from 13-inch to 30-inch swing; of patternmakers' lathes in several sizes and of upright drill presses from 20-inch swing to 50-inch swing. The catalogue is handsomely illustrated.

FAY & SCOTT, Dexter, Me. Catalogue of patternmakers' lathes which are made in a full line of sizes from a 90-inch faceplate lathe containing new features to a 10-inch lathe for manual training schools. There is also a 36-60-inch extension gap lathe illustrated.

"Air Power" is the title of a new quarterly issued by the Rand Drill Co., of New York. It is an attractive 24-page publication, primarily in the interests of the Rand Drill Co., and containing an abundance of readable and valuable material, fully illustrated, upon subjects pertaining to compressed air and its use.

WILMARTH & MORMAN CO., Grand Rapids, Mich. The "New Yankee" Drill Grinder and Other Tools, Catalogue No. 90. This describes and illustrates the company's "New Yankee" Drill Grinder, Friction Countershafts, Arbor Presses, and the "Nelson Loose" Pulley. A description of the various styles is given.

THE DERRY-COLLARD CO., 256 Broadway, New York, have started a "Book News Service" consisting of pamphlets issued from time to